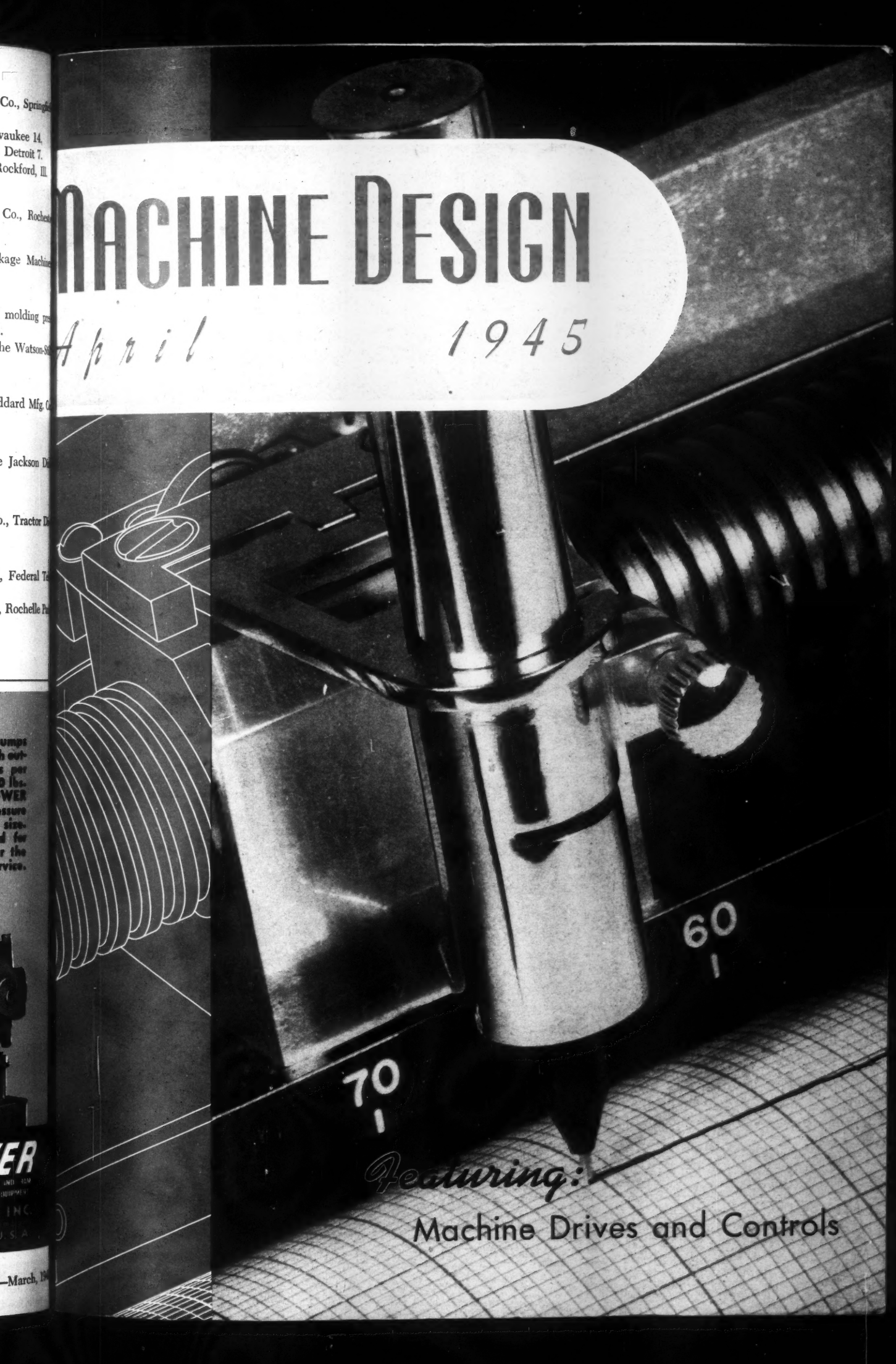
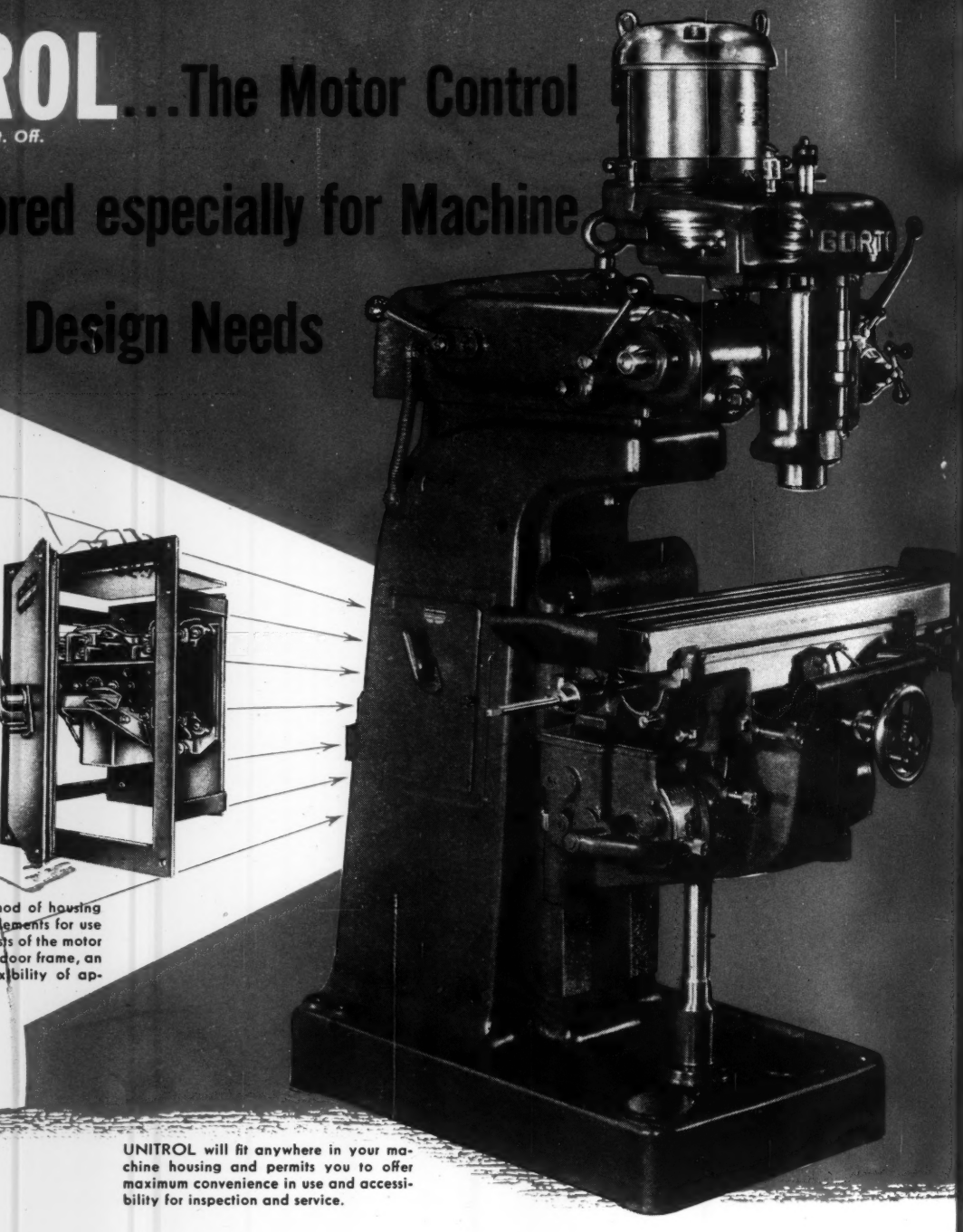

MACHINE DESIGN
April 1945
60
70
Featuring:
Machine Drives and Controls

UNITROL...The Motor Control

T. M. Reg. U. S. Pat. Off.

that's tailored especially for Machine Design Needs

UNITROL is a better method of housing starters and other control elements for use in machine design. It consists of the motor control and the UNITROL door frame, an assembly with extreme flexibility of application.



UNITROL will fit anywhere in your machine housing and permits you to offer maximum convenience in use and accessibility for inspection and service.

The advantages of UNITROL-for-machine-design are numerous and notable:

- √ New freedom of design . . . control becomes virtually a part of the machine housing, offering utmost convenience to operator.
- √ Simplifies assembly . . . shortens time and labor in mounting and wiring.
- √ Streamlined . . . no bulky control elements *outside* the machine.
- √ Saves material costs . . . no *inside* mounting base, no *outside* enclosing case.
- √ Extreme accessibility for servicing and inspection . . . parts are *out in the open.*

UNITROL, an outstanding contribution to the progress of Motor Control, beckons to every alert designer . . . points the way to *better machines* at *lower costs.* Act NOW to get this outstanding Motor Control for *your* machine by writing for complete information *TODAY.* CUTLER-HAMMER, Inc., 1310 St. Paul Ave., Milwaukee 1, Wis. *Associate:* Canadian Cutler-Hammer, Ltd., Toronto, Ont.

CUTLER-HAMMER

UNITROL

C-H

Engineering excellence finds its greatest reward in the respect and confidence of those it serves

MACHINE DESIGN

THE PROFESSIONAL JOURNAL OF CHIEF ENGINEERS AND DESIGNERS

APRIL, 1945 Volume 17—Number 4

Contents

Cover—Spectrophotometer Recording Unit (Courtesy—General Electric Co.)

Itemized Index 7

Topics 100

Forecasting Aircraft Propulsion 103
By Colin Carmichael

Basic Controls for Sequence Operations 109
By R. O. Perrine

Hydraulic Transmission Facilitates Feed Control 115
By R. L. Rougemont

Simplicity Features Intermittent Manual Drive 116
By C. W. Sweetser

Dependability, Reduced Load Achieved with Heaters 117
By Frank W. Stellwagen

Novel Applications of Roller Chain 118

Servomechanisms—How They Work 119
By R. H. Rogers

Will Aircraft Switch to 400-Cycle Auxiliaries? 125
By J. D. Miner Jr.

Where Deflection by Pressure Is Required 131
By Wm. F. Lasser

Single-Line Air Cylinder Control 134
By Spencer B. Maurer

How Planetary Reducer Aids Compact Design 135
By M. E. Cushman

Steam Turbines Drive New Locomotive 141
By Charles Kerr Jr., J. S. Newton and W. A. Brecht

Electronics Insures Accurate Loop Control 143
By L. U. C. Kelling

Selecting Drives for Speed Control—Part I—Mechanical Drives 147
By E. L. Schwarz-Kast

Look to the Airplane! (Editorial) 151

America's Armada of the Skies 152

Scanning the Field for Ideas 156

Calculating Torsional Vibration—Part I—(Data Sheet) 159
By Robert H. Scanlan

Wool Felts, Standard S.A.E. Grades (Materials Work Sheet) . . . 163

Professional Viewpoints 167

Assets to a Bookcase 168

Noteworthy Patents 169

Water Brake Simplifies Engine Control 170
By Earle S. Burnett

New Parts, Materials and Equipment 176

Men of Machines 186

Business Announcements 194

New Machines 198

Helpful Literature 385

Editor: Laurence E. Jermy

ociate Editors: John W. Greve Colin Carmichael
Richard K. Lotz

sistant Editors: Roger W. Bolz H. N. Goga

Art Editor: Frank H. Burgess

K. Price, L. E. Browne, *New York;* Erle F. Ross *icago;* R. L. Hartford, *Pittsburgh;* A. H. Allen, *roit;* L. M. Lamm, *Washington;* V. Delport, *London*

Business Staff

O. Hays, Business Manager.....Cleveland
H. Dreyer, Western Manager.....Chicago
L. Smith, Eastern Manager.....New York
L. Lawson, Asst. Eastern Manager.....New York
B. Veith, Central-Western Manager.....Cleveland
Fuller, Pacific Coast Manager.....Los Angeles
L. Callahan, Advertising Service.....Cleveland

IN OFFICE: The Penton Publishing Co., Penton Building, Cleveland 13.

NCH OFFICES: *New York* 17, 16 East 43rd St., *cago* 11, 520 N. Michigan Ave.; *Pittsburgh* 19, *yers* Building; *Detroit* 2, 6560 Cass Ave.; *shington* 4, National Press Building; *Los geles* 4, 130 North New Hampshire Ave.; *don S.W.* 1, 2 Caxton St., Westminster.

LISHED BY The Penton Publishing Co. E. L. *ner,* Pres. and Treas.; G. O. Hays, Vice Pres. and *.* Mgr.; R. C. Jaenke, Vice Pres.; F. G. Steinebach, Pres. and Secy.; E. L. Werner, Asst. Treas. Pub- *ed* on seventh of month. Subscription in U.S. and *sessions,* Canada, Cuba, Mexico, Central and South *erica:* Two years, $10; one year, $6. Single copies, *nts.* Other countries: Two years, $14; one year, $8.

yright 1945 by The Penton Publishing Co.

Better Porcelain Enameled Products from Inland Research

Ti-Namel—The New Alloy Steel Base for Vitreous Enamel Also Lowers Cost of Product

For many years the Inland research staff has been studying and experimenting toward the development of a better base for porcelain enamel—a base that would simplify operations, reduce shop time and labor costs, and produce a superior product. The result of this intensive research is Inland Ti-Namel—the new titanium alloy steel.

During the research period Inland Metallurgists worked on almost every possible combination of alloy. Finally it was determined that titanium would combine with the carbon in the steel to form a sufficiently stable carbide which is essential for the successful application of a thin white cover coat or coats to a base material without the necessity of a ground coat. Then followed a long series of tests to establish the amount of the alloy needed and the process to be used in making this titanium steel. Finally open hearth tests were made and the steel was sent to enameling shops for actual tests in making commercial products. Not until all this preliminary work was completed did Inland announce Ti-Namel—the superior alloy steel base for better porcelain enameled products.

Pending patent applications on the new enameling process and product made thereby are owned jointly by Inland Steel Company and The Titanium Alloy Manufacturing Company under trust agreement.

We have a new descriptive bulletin on Ti-Namel and will be glad to send you a copy.

INLAND STEEL COMPANY

38 South Dearborn St., Chicago 3, Ill.

Sales Offices: **Cincinnati • Detroit • Indianapolis • Kansas City • Milwaukee • New York • St. Louis • St. Paul**

INLAND TI-NAMEL

Itemized Index

Classified for Convenience when Studying Specific Design Problems

sign Calculations:
Torsional vibration calculations improved, Edit. 159-162

sign Problems:
400-cycle auxiliaries for aircraft, Edit. 125-130, 190
Aircraft propulsion forecasting, Edit. 103-108
Bellows control, selecting of, Edit. 131-133
Drying sheet facilitated with trapped air, Edit. 156
Electronics insures accurate loop control, Edit. 143-146, 172-174
Engine control utilizes variable V-belt drive, Edit. 156-157
Heaters reduce motor load, Edit. 117, 170
Hydraulic transmission for feed control, Edit. 115, 116
Intermittent manual drive, Edit. 116, 117
Planetary reducer aids compact design, Edit. 135-140
Pneumatic single line control, Edit 134
Sequence operation, basic controls for, Edit. 109-114
Servomechanisms, how they work, Edit. 119-124
Turbine locomotive described, Edit. 141-142, 192
Variable speed drives, selecting of, Edit. 147-150
Water brake simplifies control, Edit. 170

ngineering Department:
Design service, Adv. 201, 227, 242, 307, 311, 338
Equipment and supplies, Adv. 85, 198, 210, 230, 257, 292, 378, 383, 393, 402, 411
Instruments, testing, Adv. 280, 327

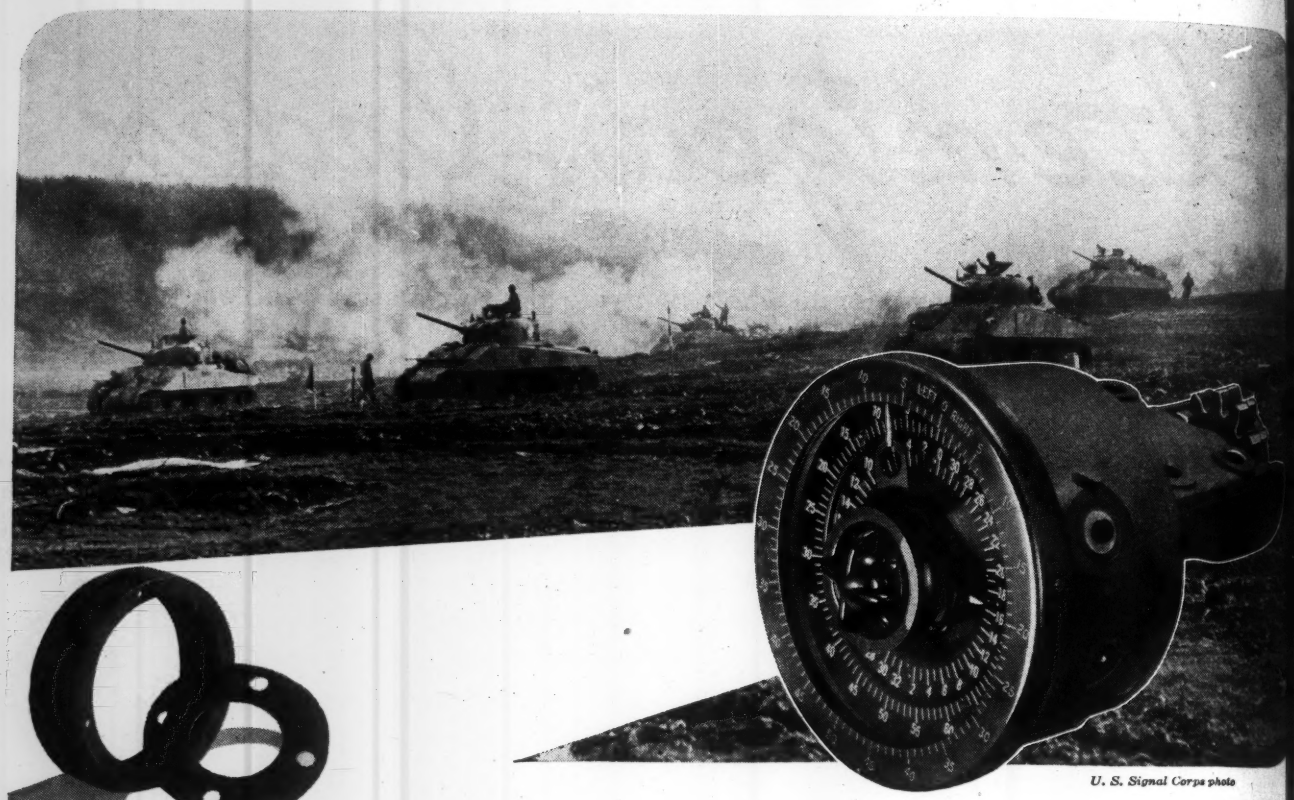
nishes:
Corrosion resistant, Adv. 413
Enamel, Adv. 215
Paints, Adv. 74, 75

aterials:
Aluminum alloys, Adv. 218, 219, 253, 276, 281
Babbitt, Adv. 384
Brass, Adv. 249, 250
Bronze, Adv. 351, 352, 396
Carbon, Adv. 231
Felt, Edit. 163-166; Adv. 76, 275
Fiber, Adv. 247
Glass, Adv. 258
Insulation, Adv. 314
Magnesium alloys, Adv. 40, 41, 347
Molybdenum, Adv. 197
Nickel alloys, Adv. 79, 226, 239
Plastics, Edit. 158; Adv. 34, 43, 45, 51, 52, 55, 69, 173, 185, 211
Powder metal, Adv. 204, 254
Rubber and synthetics, Adv. 8, 87, 90, 273, 349
Steel, Adv. 4, 26, 27, 59, 177, 181, 278, 305, 315, 388
Tungsten alloys, Adv. 16, 17, 309
Wood and plywood, Adv. 341

rts:
Balls, Adv. 348, 393
Bearings, Edit. 157-158, 167, 169, 176; Adv. 6, 12, 14, 28, 57, 171, 199, 205, 217, 229, 241, 264, 266, 269, 286, 294, 299, 391, 407, 409, BC
Bellows, Edit. 131, 133; Adv. 77, 287
Belts, Adv. 282, 359, 377
Blowers, Edit. 182
Brakes, Edit. 170; Adv. 368
Brushes, Adv. 256
Brushes, carbon, Adv. 328
Cable controls, Edit. 116, 117; Adv. 203, 362, 391
Cams, Adv. 404
Cast parts, Adv. 39, 48, 88, 187, 193, 260, 308, 333, 370
Chains, Edit. 118; Adv. 70, 72, 82, 93, 234, 277
Clutches, Edit. 176, 178; Adv. 89, 92, 345, 390, 396
Controls, electrical, Edit. 109-114, 119-124, 143-146, 172-174, 178, 180, 182; Adv. IFC, 37, 38, 47, 84, 94, 189, 220, 224, 225, 235, 244, 245, 261, 268, 270, 279, 293, 296, 317, 318, 329, 334, 342, 355, 365, 372, 374, 379, 382, 394, 397, 399, 403, 411
Counters, Adv. 56
Couplings, Adv. 22, 304, 354, 358, 398
Electrical equipment, Adv. 67
Electrical accessories, Adv. 61, 340, 402, 415
Engines, Edit. 103-108; Adv. 380, 405
Fabric parts, Adv. 373
Fastenings, Edit. 158; Adv. 1, 20, 42, 54, 96, 191, 290, 291, 310, 312, 335, 360, 362, 364, 367, 371, 388, 390, 391, 411, 418
Filters, Adv. 31, 302
Fittings, Adv. 232, 297, 348, 392, 412
Forgings, Adv. 237, 319, 326, 343, 382, 389, 398
Gears, Adv. 5, 15, 46, 49, 86, 243, 251, 298, 306, 372, 374, 378, 381, 390, 402, 404, 406, 410
Heating units, Edit. 117, 170; Adv. 405
Hydraulic equipment, Edit. 112, 113, 114, 115, 116, 120-122, 180, 182; Adv. 9, 71, 80, 97, 101, 196, 223, 238, 252, 295, 300, 301, 303, 325, 350
Joints, Adv. 19, 405
Labels, Adv. 255, 354, 383
Lubrication and lubricating equipment, Edit. 169; Adv. 23, 91, 330, 392, 396, 410
Machined parts, Adv. 346, 412
Motors, Edit. 125-130, 169, 190; Adv. 29, 60, 83, 95, 99, 195, 200, 212, 213, 228, 283, 284, 321, 338, 366, 369, 370, 389, 398, 407, 408, IBC
Mountings (rubber), Adv. 259, 339
Plastic parts, Adv. 216, 344, 363, 388, 400, 401, 417
Plugs, Adv. 214
Pneumatic equipment, Edit. 134; Adv. 102, 222, 248, 322, 332, 366, 387
Pulleys, sheaves, Adv. 44, 337
Pumps, Edit. 169, 176, 180; Adv. 33, 190, 240, 344, 346, 361, 382, 395, 400, 409, 412
Rubber parts, Adv. 320
Seals, packing, Adv. 2, 13, 58, 81, 98, 179, 183, 313, 355, 400
Shafts, flexible, Adv. 184, 368, 370, 380
Sheet-metal parts, Adv. 62, 406, 409
Speed reducers, Edit. 135-140; Adv. 10, 11, 175, 206, 207, 209, 272, 357, 404
Springs, Adv. 30, 73, 221, 271, 331, 342, 394, 406, 410
Stampings, Adv. 32, 53, 324, 360
Transmissions (variable speed), Edit. 115, 116, 147-150, 156-157; Adv. 246, 372, 415
Tubing, metallic, Adv. 63, 64, 65, 66, 208, 262, 263, 265, 267, 274, 336, 389
Tubing nonmetallic, Edit. 182; Adv. 194, 376
Universal joints, Adv. 202, 356, 378, 394
Valves, Adv. 35, 192
Welded parts and equipment, Adv. 24, 25, 285, 358, 393
Wheels, casters, Adv. 340, 392
Wire, Adv. 68, 353, 364
Wire cloth, Adv. 21, 316

Production:
Grinding, Edit. 158; Adv. 78, 288, 289
Hardening, Adv. 36, 236
Instruments, Adv. 374
Service facilities, Adv. 407
Tools, Adv. 18, 233, 323, 375, 376, 380, 413

MACHINE DESIGN is indexed in Industrial Arts Index and Engineering Index Service, both available in libraries generally.



U. S. Signal Corps photo

The Aim is... ACCURACY!

Tanks and tank destroyers now serve a dual function . . . for close range fire-power and as mobile, armored artillery units with an effective range of several miles.

A precision aiming device . . . called the *AZIMUTH INDICATOR* . . . enables the crews of tanks and tank destroyers to quickly adjust a gun turret to a direct horizontal and, with the aid of reconnaissance planes, the device accurately and quickly courses the aiming for the transverse position.

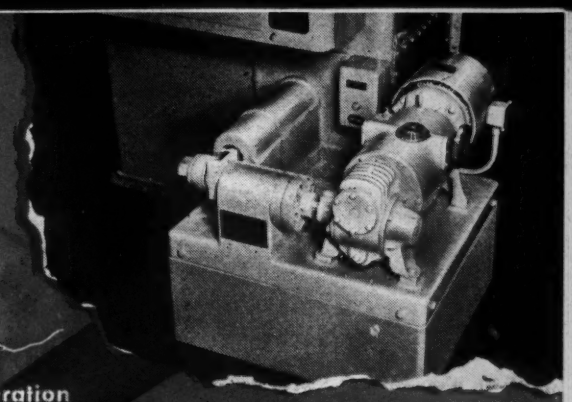
The *azimuth indicator* (illustrated) is made by the Beam Manufacturing Company of Webster City, Iowa. The rubber parts (illustrated) are produced for "Beam" by The Ohio Rubber Company. Like all components of the *azimuth indicator*, these rubber parts must conform to rigid standards.

From beginning of manufacture to "end use" of the *azimuth indicator*, the aim is ACCURACY. When *YOUR* aim is accuracy in mechanical molded and extruded rubber or synthetic rubber parts—call for—

"ORCO-OPERATION"

THE OHIO RUBBER COMPANY · WILLOUGHBY, OHIO

BRANCHES: DETROIT · NEW YORK · CHICAGO · INDIANAPOLIS · WASHINGTON · CLEVELAND



Stepless VARIABLE SPEED operation

offers tremendous advantages on many applications. You, yourself, probably have several applications on which variable speed operation would improve the quality of the product and increase the quantity of production as well.

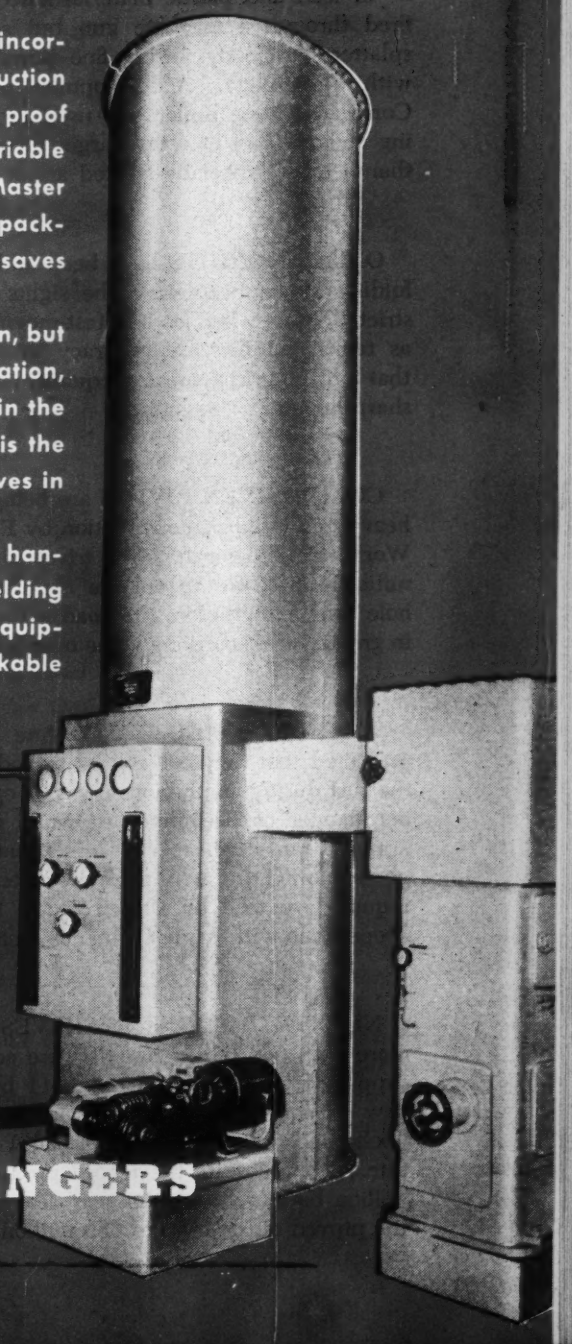
For example, look at this oxygen generator which produces from ordinary air, absolutely dry oxygen that is at least 99.5% pure. This Speedranger provides stepless variable speeds so that pure oxygen and only pure oxygen is pumped off. Provides a high uniform quality product at peak production rate.

But look more closely at this variable speed drive. It incorporates not only the variable speed unit, but also a gear reduction to give speeds in exactly the right range, and an explosion proof motor. All of these . . . the explosion proof motor, the variable speed unit, and the gear reduction . . . all are standard Master units that easily combine into one compact, integral power package. Saves ordering and mounting time . . . saves space . . . saves money.

Probably you will not need exactly this same combination, but Master Speedrangers are available for every current specification, every type enclosure, with gearheads, unibrakes . . . and in the whole wide range of types in the Master line . . . in fact, is the most flexible, the most versatile line of variable speed drives in the world.

The next time you need a drive for material processing, handling, and conveying equipment; mixers and agitators; welding positioners; machine tool drives; testing and calibrating equipment . . . to name only a few . . . see what a really remarkable job Master Speedrangers can do for you.

THE MASTER ELECTRIC COMPANY
DAYTON 1, OHIO



MASTER SPEEDRANGERS

FRANGIBLE BULLETS consist of a combination of lead and plastic material hard enough to be fired through a machine gun but soft enough to splatter harmlessly into a fine powder on contact with armor plate. A development of the Training Command, these bullets are used for combat training and are shot at a fast target plane having more than a ton of specially treated armor plate.

OPTICAL SIGHTS have been added to the new folding model bazooka. The sights provide unrestricted vision when leading fast-moving targets such as tanks and increase accuracy to such an extent that an infantryman may qualify as a "bazooka sharpshooter".

COLD-DRIVEN RIVETS are being employed on heavy mill building construction by Fort Pitt Bridge Works of Pittsburgh. The rivet is driven under sufficient pressure to enlarge the shank to fill the hole completely before the head is formed, resulting in greater stress-carrying value over hot-driven rivets.

WATERPROOFED TANKS are so thoroughly protected that they can roll off a landing barge into the surf during amphibious invasion operations without danger of becoming "drowned out". Heavily armored tanks can operate in six feet of water and go ashore fighting with this new waterproofing technique, developed by United States Rubber Co. in cooperation with Army Ordnance technicians.

NEW PETROLEUM RESERVES amounting to more than two billion barrels were added to the oil supply of the nation during 1944 by discovery of new oil pools and the further development of older pools, according to the American Petroleum Institute. This, despite the record production of 1,678 million barrels during the year, leaves a net gain in the proved oil reserves of 389 million barrels.

LOW - TENSION IGNITION system for aircraft has been developed by Scintilla Magneto division of Bendix to make practical flights at altitudes of 50,000 feet and over. The new system makes possible higher voltage sparks at the spark plugs while replacing a high-tension current of more than 12,000 volts with a low-tension current. Better engine performance, easier control of electric current, less radio interference, and increased resistance to moisture and other atmospheric conditions are achieved with the system.

VISCOSITY INDEX IMPROVER, a new organic compound known as Acryloid HF developed by Rohm & Haas, when added to hydraulic and recoil oils prevents dangerous thickening at subzero temperatures or excessive thinning in tropical heat.

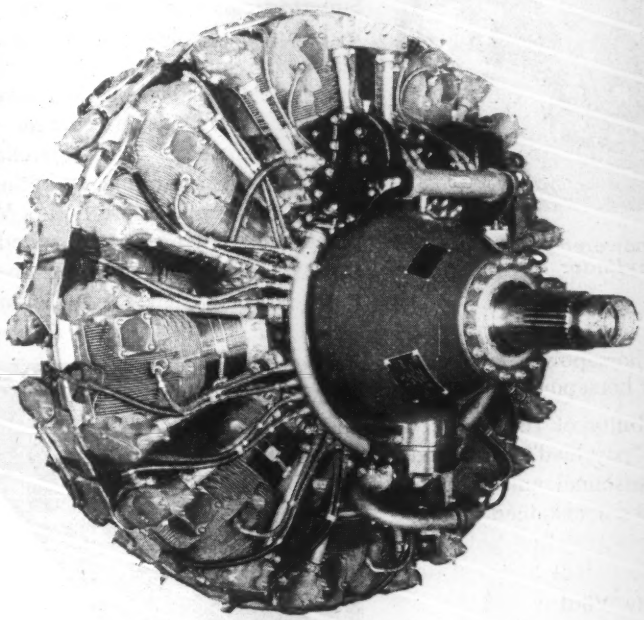
PRODUCTION DIFFICULTIES, manpower shortages and increasing military demands for copper, steel and aluminum have resulted in a tighter controlled materials situation for the second quarter of 1945 than at any time during the last several quarters.

TO AID OBSERVERS in "Hell's Acre"—a mammoth new hangar at the Army's southern proving grounds for subjecting airplanes to temperature ranges from −70 to 165 degrees—tiny self-propelled cars completely enclosed in glass and resembling a Buck Rogers device will carry their own miniature air-conditioning systems to protect the men against heat exhaustion.

DEMANDS ON FACILITIES of pulp and paper machinery manufacturers for production of military equipment have delayed production of new mill machinery amounting to more than nine and one-half million dollars, according to WPB.

SEVEN-CYLINDER Wright cyclone aircraft engine has been developed to produce 700 horsepower on inexpensive low-octane fuel. The new engine is provided with a 2-speed supercharger drive.

SURPLUS PROPERTY BOARD has announced that 7,013 civilian type planes, out of a total of 10,180 declared surplus, have been sold and paid for as of February 1.



Forecasting Aircraft Propulsion

By Colin Carmichael
Associate Editor, Machine Design

TO MEET the propulsion requirements of present-day and future aircraft is a task which is taxing to the limit the resources of powerplant engineers. Specifications which are calling for speeds approaching that of sound, altitude performance reaching into the stratosphere, and enormous total horsepowers for new giant long-distance bombers and transports have tended to outstrip developments in the means of propulsion. As a result the whole thinking of aircraft engineers is in a state of flux such as has not existed since the Wright brothers successfully demonstrated the eminent suitability of the gasoline engine and propeller as a driving medium.

Because many engineers in other fields take more than passing interest in aircraft developments, it is hoped that this article will stimulate thought on the fundamental problems of aircraft propulsion. Recent engineering trends will be discussed, and the significance of new developments such as the gas turbine, jet propulsion and rocket propulsion, as well as the conventional engine-propeller arrangement, will be evaluated in terms of their suitability in particular fields of application.

Aircraft propulsion means of whatever type must be judged primarily on the

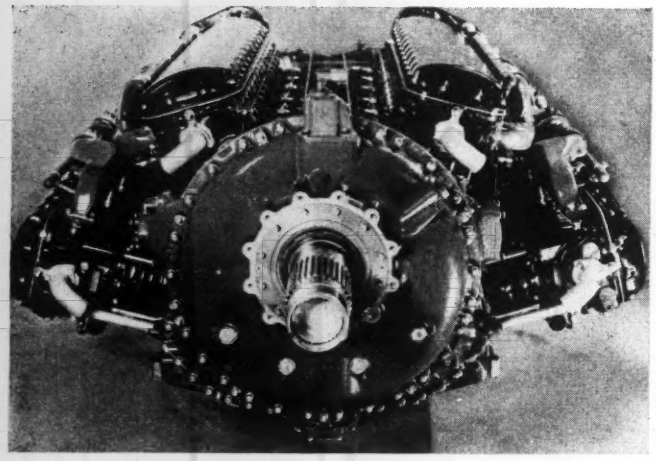


Fig. 1—Representative of high-powered liquid-cooled engines is the Allison 24-cylinder engine

basis of the following two characteristics:

a. Weight of complete plant per horsepower
b. Weight of fuel consumed per horsepower per hour.

Upon these factors depends the ability of the complete plane to take off with a reasonable "pay load" which may include bombs and ammunition, personnel and supplies, passengers and freight, etc., and make a sustained flight at the required speed.

Air-Handling Capacity Vital

In aircraft which depend on the atmosphere for oxygen to support combustion of the fuel, the horsepower which can be developed is directly proportional to the engine's ability to consume air. Hence the weight of an engine is determined by the amount of equipment necessary to handle that air, mix it with fuel, burn it and convert the released energy into usable power. Thermal efficiency of the conversion also is a significant factor in determining the size of the engine required as well as the weight of fuel to be carried.

Secondary factors whose importance at present depends upon the intended service of the plane are

c. Ability to perform at high altitude
d. Ability to furnish high plane speed.

While high-altitude flying frequently is associated more with avoiding flak or bad weather than with economics, the reduced drag in the rarefied upper air has an important effect on fuel consumption and therefore on take-off weight as well as upon speed. Super speeds, now thought of as mainly a military requirement, may eventually become standard practice for all except certain special-purpose ships.

In order to see the more radical developments in their true perspective, the present status and future possibilities of the conventional engine-propeller system will first be discussed.

RECIPROCATING ENGINES: Present-day reciprocating engines for high power develop approximately 125 horsepower per cylinder at take-off and have thermal efficiency at cruising power exceeding 30 per cent. Cylinder bore and stroke are approximately 6 inches and the speed ab 2500 to 3000 revolutions per minute. Weight is clos one pound per take-off horsepower. By providing enou cylinders, comparatively large total horsepowers can developed. Thus the 18-cylinder Wright engine shown the head illustration delivers about 2200 horsepower, wh the 24-cylinder Allison engine, *Fig.* 1, is rated at 2 horsepower. Four of the Wright engines power each B Superfortress, while the giant B-19 recently had the or inal engines replaced with a set of the large Allison e gines.

High output both at sea level and altitude is aid through use of a supercharger which increases the a handling capacity. The supercharger may be gear-dri or driven by a turbine operating from the exhaust g High-altitude planes use both. With normal boost the s percharger delivers to the cylinders air at approximate sea-level density even when operating at high altitud Extra power for take-off and climb results from permitti

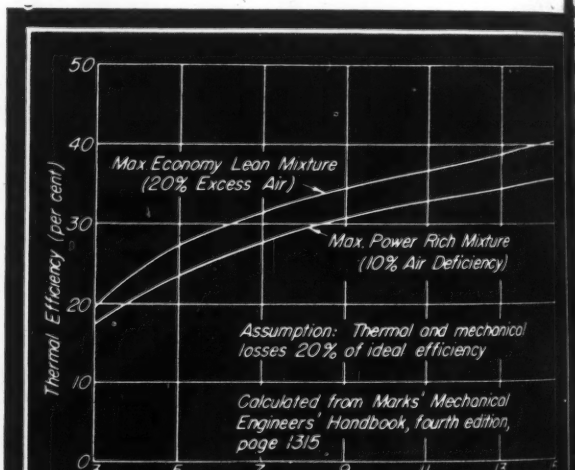


Fig. 2—How thermal efficiency of gasoline engines varies with compression ratio and fuel-air mixture

the supercharger to overboost and deliver air to the cyli ders at higher pressure.

Development of substantially higher horsepower pe engine hinges on increasing the capacity to handle ai Four principal ways of doing this are:

a. More cylinders per engine
b. Larger cylinders
c. Higher revolutions per minute
d. Greater supercharge boost.

Improved thermal efficiency also results in more powe as well as better economy and less fuel weight. Followin are some of the problems that must be solved in order t effect increased horsepower performance by these means.

MORE CYLINDERS: As the number of cylinders per en gine is increased the mechanical complications multiply while in the case of air-cooled engines the cooling problem becomes acute. It is no secret, for instance, that cooling of the 18-cylinder double-row radial engines in the B-29

a major problem that had to be licked before the spec-lar performance of this ship could be realized. To in-se the number of cylinders above 18 will involve addi-al rows. For example, larger air-cooled engines with cylinders are under development. With a fair average 25 horsepower per cylinder, it is conceivable that such engine could put out $28 \times 125 = 3500$ horsepower, or re, when fully developed.

With liquid-cooled engines there seems to be a practical it to the number of cylinders per bank as well as to number of banks that can be accommodated; the 2600 sepower Allison engine, *Fig.* 1, has four banks of six inders each. Additional cylinders per bank accentuate vibration problem.

Why Not Bigger Cylinders?

LARGER CYLINDERS: Although larger-bore cylinders uld seem to offer a promising means of increasing horse-wer output per cylinder, several obstacles stand in the y of further development in size. One is the increased fficulty of cooling the cylinders, valves and pistons; with more than one-third of the heat energy in the fuel con-ted to work, two-thirds must be dissipated as heat. In-much as the energy released is proportional to the volume the cylinder, which varies as the cube of the bore, while surface for dissipating heat is proportional to the square the bore, it will be appreciated that cooling difficulties rease in direct proportion to an increase in bore. While id cooling can remove heat more effectively from the nder itself, the question of piston and valve cooling remains.

Another difficulty is that the weight per horsepower ds to increase with bore. If a new engine could be olved from an existing design by increasing every di-nsion in the same proportion, all stresses, including in-ia stresses, would be identical if the pressures in the nder and the piston speed remained the same. Under se conditions the horsepower of the larger engine would rease as the square of the bore while the weight would rease as the cube of the bore; the weight per horse-wer therefore would increase directly as the bore. A rd factor is the resulting increase in frontal area pre-nted by the whole engine as the cylinder size is in-eased. It would seem that cylinder size is not likely to increase more than a few per cent.

HIGH SPEED: Increasing the revolutions per minute is subject to two major limitations: One is the increase in the inertia stresses, which means heavier moving parts, and the other is the increasing difficulty of getting the fresh charge into the cylinder through the limited opening offered by the valves. While speeds may increase somewhat, such increase probably will be limited to a few per cent.

SUPERCHARGING: Increased power through greater supercharge within the limitations of available fuels is more promising. Two factors are involved: Engine knock (detonation) and cooling. Engine knock is the result of squeezing and heating the air-fuel mixture until it acquires the properties of a high explosive. In normal combustion a flame initiated at the spark plug or plugs travels across the cylinders at a finite speed less than the speed of sound and is accompanied by a relatively smooth pressure rise. When detonation occurs, however, that part of the mixture which has not yet burned explodes with shattering violence. Compression of this unburned mixture to the detonating point occurs in three stages: In the supercharger, in the cylinder during compression stroke and in the combustion chamber by the flame itself. Occurrence of detonation may be delayed by proper design and cooling of the cylinder head and pistons, and by the use of high-octane fuel. High-octane fuels increase the power by permitting higher degrees of supercharge (to increase air capacity) and higher compression ratios (to raise thermal efficiency). Contrary to popular opinion such fuels have no more inherent power, the heat value being about the same for all gasolines regardless of octane number.

What Can Be Done with Super Fuels

Recent developments in fuels indicate that knock need no longer be a limiting factor in the development of maximum power. However, for the reason that heat to be dissipated is roughly proportional to power developed, the cooling problem is vital in any attempt to increase power through supercharging.

According to performance figures obtained on a liquid-cooled test engine using a new super fuel, the present 125 horsepower per cylinder on 100-octane fuel might some day be increased to as much as 330 horsepower. Such a gain would make it possible in a 24-cylinder liquid-

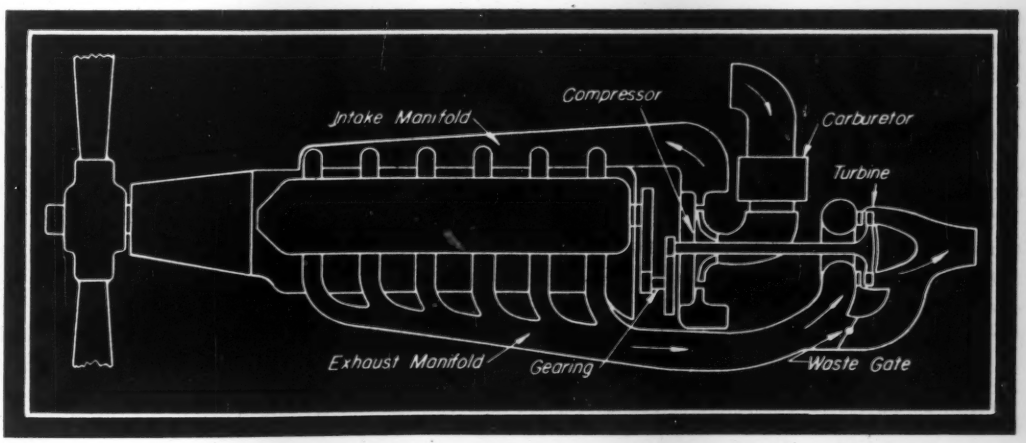


Fig. 3—Compound engine utilizes exhaust energy to drive turbine which is geared to crankshaft as well as driving supercharger

cooled engine of present-day design to develop as much as 8000 horsepower.

THERMAL EFFICIENCY is a function of the thermodynamic cycle upon which the engine operates. Primary factors influencing thermal efficiency are the maximum and minimum temperatures of the cycle—in general, the greater the extremes the higher the efficiency. In the reciprocating engine relatively high thermal efficiencies are possible because, due to the intermittent occurrence of the maximum temperature, the cylinder, piston and valve materials operate at temperatures far below the maximum with the result that combustion temperatures of 4000 degrees Fahr. or more may be employed. Aiding the attainment of high temperature ratio is compression ratio, one

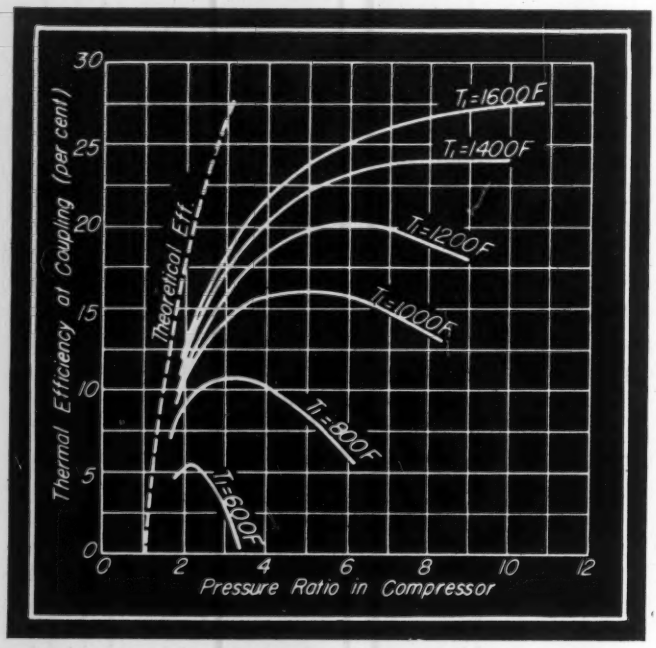


Fig. 4—Thermal efficiency of gas turbine or turbojet engine varies with maximum cycle temperature and with pressure ratio, as shown on this Allis-Chalmers chart

of the most important design factors affecting thermal efficiency, *Fig.* 2. As compression ratios increase beyond a certain point, however, the mechanical efficiency tends to drop off due to the heavier moving parts needed. Present aircraft compression ratios are about 7:1 or less, while optimum compression ratio, assuming a suitable antiknock fuel, seems to be about 10:1. Increasing the ratio from 7 to 10:1 would result in about 13 per cent improvement in thermal efficiency. Further, because more of the heat is converted into useful work, less heat remains to be dissipated by cooling and the engine could be expected to carry a higher supercharge.

Some additional power may be obtained from conventional engines by other means. For example, the exhaust from the engines or from the turbosupercharger may be concentrated in the form of a jet to provide additional thrust. This feature is being incorporated in new designs.

Again, the injection of a water spray into the inlet to aid cooling and suppress detonation permits considerable increase in output, especially in emergencies. Further, it is possible to design the cowling around an air-cooled engine so that the heated air furnishes a jet to provide a thrust which more than offsets engine drag.

COMPOUNDING: A possible line of development now under investigation is to employ an exhaust gas turbine larger than would be required for supercharging alone, geared to the engine shaft, *Fig.* 3†. The combination is, in effect, a compound engine in which the cylinder is the high-pressure unit and the turbine the low-pressure unit. By increasing the exhaust back pressure, a larger proportion of the total power will be developed in the turbine. The turbine can extract more power by expanding the exhaust gas to atmospheric pressure, which the limited-volume reciprocating engine cannot do. It has been estimated that by this means the thermal efficiency at cruising speed might be raised to 40 per cent, with a reduction in weight per horsepower of about 12 per cent†.

Problems in Applying the Diesel

DIESEL ENGINES: Comparing the diesel at its present stage of development with the gasoline engine, the weight per horsepower is greater due to heavier construction and lower operating speed, while thermal efficiency is slightly better and the less volatile fuel is less of a fire hazard. Because the heat of compression is relied upon to initiate combustion, difficulty is experienced in obtaining prompt combustion at high speed due to the "hang-fire" characteristic of most fuels. As a result an accumulation of injected fuel in the cylinder suddenly inflames, causing diesel knock. Fuel of high "cetane" number (which is an opposite characteristic to octane number) must be used and a high compression ratio employed.

So far, diesel engines for aircraft have been limited in size to units of less than 1000 horsepower. However, high supercharge hastens combustion rather than otherwise and therefore tends to suppress knock, so there is prospect of further development although at the present time there is little activity in this field.

Limitations of Present-Day Propellers

PROPELLERS: Conventional types of propellers are subject to serious limitations when their tip speed approaches the speed of sound. Because local air speeds over the blade necessarily must exceed the average speed in order to develop a thrust, actual propeller speed must remain less than the speed of sound if "compressibility burble" and resulting loss of efficiency is to be avoided. For plane speeds much in excess of 450 miles per hour, therefore, the conventional propeller offers little promise. Further, for the absorption of large amounts of power a large diameter propeller is necessary and the revolutions per minute must be kept low to avoid too high a tip speed. In recent large installations the speed at cruising is only 600 revolutions per minute, reduction gearing being employed between the engine and propeller. For propellers of similar design working with the same tip speed, the weight of the propeller per horsepower increases directly as the diameter

†C. S. Bachle—"Some Possibilities of Turbine Compounding with the Piston Engine", S.A.E. paper presented at annual meeting, January, 1945.

ersely as the revolutions per minute. Multiple pro-llers, either side by side or in tandem (dual rotation), y be employed to keep the weight down where large wer must be absorbed. With such a system the engine y be located at any convenient point, perhaps inside fuselage, the power being transmitted to the propellers her through mechanical gearing and shafting or by ctrical means.

Reciprocating Engine Will Continue

Conventional engine-propeller systems, having been ought to a high state of mechanical perfection and with spects of further improvement, may be expected to ntinue serving for some years in many types of planes, rticularly where moderately high speed and good fuel nomy are important. Radical new powerplants on ich intensive development work is being done may over-n older ideas of what constitutes high speed, and offer exciting prospect. The more significant of these will discussed in the light of their potential applications.

JET PROPULSION: Two jet-propelled U. S. fighter planes eady announced, the Bell P-59 "Airacomet" and the ckheed P-80 "Shooting Star," employ the type of power-nt technically known as the "turbojet," which was dis-sed in a previous article†. Here the breakaway from nvention is complete, and the performance of these nes places them in a new category. The P-80, for ex-ple, is said to fly faster than any plane hitherto de-loped. The turbojet unit in this craft is installed in the selage between the cockpit and the tail. It is far more werful than any conventional reciprocating engine for craft.

However, the new powerplant does consume more el per horsepower per hour, due primarily to the fact that

TABLE I*

Powerplant Plus Fuel Weights Compared

Type of Powerplant	Powerplant Weight per Cruising hp	Thermal Efficiency Per Cent	Propulsive Efficiency Per Cent	Total Weight Lb per Trip per Thrust hp
For 300-mile trip at 550 mph				
rbojet	.51	16	64	.9
rboprop eller	1.1	16	52	3.1
mpound	3.0	40	52	5.9
soline†	3.4	31	52	6.9
el	3.9	40	52	7.8
For 3000-mile trip at 300 mph				
mpound	3.0	40	83	8.6
el	3.9	40	83	9.7
soline†	3.4	31	83	10.5
rboprop eller	1.1	16	83	14.0
rbojet	.51	16	40	26.5

*From S.A.E. paper by C. S. Bachle.
†Turbosupercharged.

operates on a continuous cycle, which does not permit aximum temperature to exceed that at which the metal rts of the turbine can operate. Propulsive efficiency, hich is a function of plane speed to jet speed ratio, is low cept at the highest speeds because of the large percent-ge of jet energy thrown away in the slipstream. With gher operating temperatures the thermal efficiency and

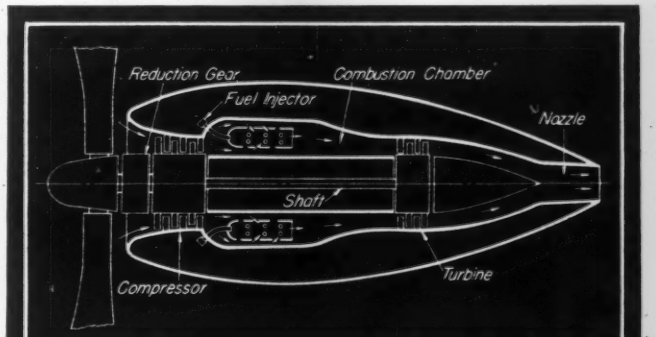
†"Analyzing the Jet Propulsion Engine", MACHINE DESIGN, July, 1944.

jet speed will increase as shown in *Fig.* 4, but propulsive efficiency will drop unless plane speed also increases.

At high altitude the performance improves because the extremely low air temperature improves the thermal efficiency while the low air resistance permits the attainment of sufficiently high speed. At high speed the turbojet is greatly aided by the "ram" effect, or conversion of the kinetic energy in the high-velocity air entering the compressor into potential or pressure energy. This increases the overall effective pressure ratio, thus improving the efficiency (*Fig.* 4) and output power.

Considerable emphasis has been placed on the simplicity of the turbojet engine and the rapidity with which

Fig. 5—Below—Schematic diagram shows how gas turbine may be used to drive a propeller as well as furnishing a jet



it may be replaced in a plane. It may be reasonable to conclude that a new concept of endurance has arisen, whereby the unit or certain parts of it are considered "expendable" and therefore are operated at temperatures which permit high performance but, at the same time, relatively short operating life.

GAS TURBINES: To maintain high thermal efficiency and to improve the propulsive efficiency would involve extracting more energy from the turbine in order to reduce jet speed. This can be done by increasing the size of the turbine and absorbing the additional power by means of a propeller geared to the turbine shaft, *Fig.* 5. The resulting combination is a "turbopropeller" drive in which the propeller is primary and the jet secondary. Turbine efficiency, which depends on ratio of bucket speed to jet speed, is independent of plane speed and therefore can be reasonably high even on a moderate-speed plane. Propeller efficiency also is higher than jet-propulsive efficiency at moderate speeds. Present indications are that the gas turbine-propeller drive will find wider application in the near future than the turbojet, which will be limited to superspeed planes. A favorable aspect is the absence of stringent fuel specifications, as either type will burn a wide variety of fuels.

Because the thermal efficiency of the gas turbine increases with size, while large single propellers offer difficulties, it has been proposed to employ in some cases one large turbine, perhaps located in the fuselage, with an electric drive to multiple propellers located in the wings. New 400-cycle generators and motors are light enough in

weight to make such a scheme feasible.

How the five major types of powerplant already discussed compare for short high-speed trips and for long moderate-speed trips is shown in TABLE I. For the short trip, powerplant weight is all-important while for the longer trip the product of propulsive and thermal efficiencies would control the choice of powerplant because of the resulting fuel economy.

Engines for "Vengeance Weapons"

BUZZ BOMBS: A form of jet-propelled plane which dispenses with the compressor and turbine is the "impulse-duct" or "resonant-tube" engine. Employed in the German V-1 flying bomb and also now in production in this country, the unit is simply a tube about ten feet long with a series of check valves at the front end and open at the rear. It operates on an intermittent cycle with a limited amount of compression resulting from the periodic surge of flame back and forth in the tube, which resonates like an organ pipe. Fuel is injected into the air as it streams past the check valves while the flame is surging to the rear; when the flame surges forward again the pressure rise closes the check valves and the mixture explodes.

Thermal efficiency is low, probably not over 2 per cent, but the unit is light in weight—about one-third of a pound per horsepower—and of inexpensive construction. It is strictly expendable and consequently appears to have application only in the form of missiles.

ROCKET PROPULSION: Known as the thermal jet engine, the type of propulsion used by rockets carries its own oxygen, thus differing fundamentally from powerplants previously considered. For aircraft propulsion, thermal jets using solid fuels such as nitroglycerine and nitrocellulose are employed to assist in take-off of flying boats and land planes and also are used to launch the buzz-bomb type of flying missile. The units develop a high thrust for a short time, which is usually measured in seconds, but once the reaction is started it cannot be controlled, hence application to sustained flight is not practical.

Liquid Fuel Permits Intermittent Operation

When liquid fuels are employed for a thermal jet, control of the jet is possible, as in the case of the German ME-163 fighter. Fuel and oxygen last only a few minutes if used continuously but actually the plane can stay in the air for a considerable time by soaring and gliding, using the thermal jet only in short spurts. Weight of the equipment is light [illegible] to the power which can be developed, the only equipment being storage tanks for fu[el], means to deliver fuel to the burners, and the combust[ion] chamber and nozzle.

Rocket projectiles, as distinguished from aircraft, [can] make much more effective use of the thermal jet. A[ir]craft which depend upon wings for support are limited [in] speed whereas a projectile can travel much faster than [the] speed of sound and can attain high propulsive efficien[cy] as the rocket speed approaches the jet speed. It is, [in] fact, possible to operate with these speeds equal, in wh[ich] case the propulsive efficiency is 100 per cent. The te[m]nical, if not military, success of the German V-2 roc[ket] bomb is an indication of what may be in store as [the] thermal jet engine is further developed.

Performance of an Actual Rocket

A spectacular achievement of World War II, the V[-2] weighs 12 tons, which is as much as a large fighter pla[ne] and has propulsion equipment capable of propelling it [to] a height of 22 miles in 71 seconds, when its velocity [ex]ceeds 3600 miles per hour. Thereafter, with power [cut] off, the projectile soars to a maximum height of over [50] miles and finally descends to earth 200 miles from [its] starting point. During the 71 seconds of power flight [13] tons of fuel are consumed and ejected, this corresponding to an average rate of thermal energy release of 1,350,000 horsepower. According to published figures the jet velocity is about 6400 feet per second, while the thrust is twice the initial weight of the projectile, or 48,000 pounds. At the end of the power flight the rocket velocity is 5280 feet per second, hence the speed ratio is about .82, which would correspond to a propulsive efficiency of over 90 per cent. Maximum thrust horsepower would be close to 460,000 and the net efficiency 34 per cent at top speed.

Rockets would seem to have their chief application for flight above the atmosphere, as in the case of the V-2 or for possible interplanetary travel. With present fuels the weight of fuel necessary to take a rocket projectile beyond the influence of the earth's gravity would have to be nearly 98 per cent of the total initial weight. When fuels are developed which utilize atomic energy, it will of course be possible to improve upon this figure greatly.

MACHINE DESIGN acknowledges with appreciation the cooperation of the Air Technical Service Command, Wright Field, Ohio, in connection with the preparation of some of the information which appears in this article. The article was approved by the War Department in Washington after modification of the original manuscript.

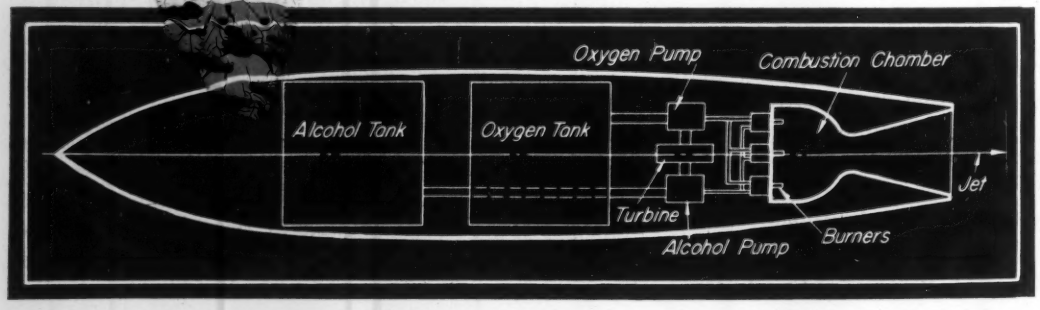
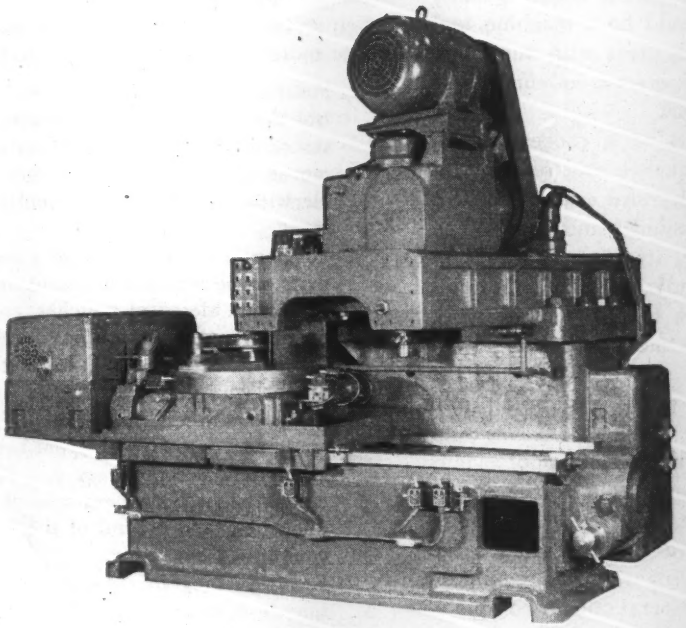


Fig. 6—A typical fo[rm] of thermal jet engine. [The] rocket engine is sho[wn] schematically at le[ft]. This type of unit [is] completely self-co[n]tained, not dependi[ng] upon the atmosphe[re] for oxygen supply.



Basic Controls for Sequence Operations

By R. O. PERRINE
Supervising Engineer
Cutler-Hammer Inc.

MANY machines and processes operate in such a manner that the electrical equipment must provide a definite sequence or program of operations. In analyzing the problems presented by such applications and designing the proper controls, there are certain basic ideas which, when recognized, form useful tools in the preparation of the complete design.

In designing an automatic machine in which electrical apparatus is to be utilized, the basic questions which must be answered before the control scheme can be developed are:

1. Will operation be fully automatic or semiautomatic?
2. What means can be used to determine the sequence of events?
3. Will there be any selective variations in the cycle and, if so, how will the variations be obtained?

Principal applications of sequence controls in industry include machines for

handling of materials and processing materials, and in machine tools. A typical example of a machine in which sequence operations are employed is shown in the head illustration.

Automatic operation when once started continues indefinitely in a repetitive manner until some action is taken to stop it. An instance of this would be a machine tool with automatic loading, or a punch press with automatic feeding which could continue its operation indefinitely as long as material or stock is available.

Semiautomatic operation consists of a process which proceeds only once through its sequence and remains inactive until again initiated manually. An example of this is a shell lathe which must be reloaded manually. When loaded and the operation initiated, its cycle proceeds in some manner as follows: The spindle starts and the tools approach the work rapidly; as the tools reach the work their speed of approach is reduced to a slow cutting feed; when the cutting is completed the spindle stops, and the tools retract rapidly to the original position, stopping until the next cycle in initiated.

In the case of some machines, the sequence of events may be entirely mechanical and connected to the condition of processing. For example, a conventional automatic screw machine has its entire sequence under the control of a master cam drum, which in one revolution mechanically performs the various operations for completely machining a single piece.

For many machines, especially those of a complica nature, a sequence may be set up electrically to grea advantage. Usually the electrical sequence results lower cost, less complication, and increased flexibility. some machines a combination of mechanical and el trical control can be used most effectively.

Sequence operations may be controlled electrically one or more of the following methods.

1. Position of material in process may be used to control the operation of the handling devices. For instance, as a pan of material on a series of conveyors moves along over the various devices it operates electrical switches which control its further movement.
2. Position of a certain part of a machine, or of several parts of the machine in combination, may be utilized to operate electrical switches, controlling the further operation. Again referring to the shell lathe example, the position of the tool feeding mechanism can set up the entire sequence.
3. Time controllers may be preferred in processing equipment so that each event happens a definite time after the preceding event. Flame-hardening machines might be an example of this. The heating, quenching and control of the flame can best be accomplished by timing.
4. Electrical program devices, in some cases, govern sequence operations entirely within the electrical system but tied up with the machine operation. Such a device might be in the form of a magnetic notching device which notches each time a machine function takes place, setting up the circuit for the next function. An electrical program device has its greatest application in two kinds of situations: 1. Where the machine cycle is completed in a different position of the machine than the position at the beginning of the cycle. 2. Where the machine cycle is variable for different type of work, and a convenient means of setup is required for establishing the sequence.
5. Combination of several of these means as well as some purely mechanical means to provide one coordinated machine and control is often found desirable in practice.

If a machine or process is operate in any manner but a de nite fixed cycle, preliminar thought must be given in dete mining how these variations ma be obtained.

When limit switches are in volved, the cycle may be varie by merely rearranging th

Features of Basic Circuits for Cycle Control

Representative Diagram	Initiating Contact	Low Voltage Protection or Release	Distinguishing Feature	Typical Application
Rotary Motions—Make One Revolution, Stop				
Fig. 1*a*	Momentary	Protection	Basic circuit	Crank-operated pusher
Fig. 1*a* (without *M* contact)	Momentary	Release	Basic circuit	Crank-operated pusher
Fig. 1*b*	Momentary or Maintained	Protection	Nonrepeat on maintained control	Punch press
Fig. 1*c*	Momentary or Maintained	Release	Nonrepeat on maintained control	Intermediate steps in a composite sequence
Fig. 1*d*	Momentary	Protection	Circuit immediately maintained	For very slow motions
Fig. 1*e*	Momentary	Release	Circuit immediately maintained	Intermediate steps in a composite sequence
Reciprocating Motions—Move Out, Back, Stop				
Fig. 2*a*	Momentary	Protection except while *FLS* limit switch is engaged	Two limit switches	Tapping machine or feed motions
Fig. 2*b*	Momentary	Protection—Forward Release—Revolution	One limit switch	Tapping machine or feed motions
Fig. 2*c*	Momentary	Complete Protection	Two limit switches	Tapping machine or feed motions
Fig. 2*d*	Momentary		Electrohydraulic maintained position valves	Feed motions
Fig. 2*e*	Momentary or Maintained		Electrohydraulic maintained position valves—nonrepeat on maintained contact	Feed motions

witches or by omitting or adding certain limit-switch functions. other situations the switches may remain fixed with electrical lector switches to commutate the circuits. Often a combination these two will be found desirable. A careful study of the varia- ns and their relation to the method of control will usually pay vidends in simplicity and flexibility of the finished machine.

The method of varying the cycle is, of course, intimately re- ted to the method of overall control of the cycle as previously scussed.

Where time is the basis for the cycle, variations can usually most readily incorporated in the time-controlling device or de- es. In the case of electrical program devices, variations may incorporated in adjustments of the program device itself or ay be incorporated in separate electrical selectors which alter e effect of the program device at any particular position.

Designing the Control Scheme

The first decision which must be made in designing a control heme for a sequence controller is the consideration of what will ke place if the cycle is interrupted for any reason whatsoever. general three situations may arise. It may be desired:

1. To resume exactly where the cycle was interrupted
2. To return each motion manually to its original position before resuming
3. To have the system biased to return automatically to its original position.

A sequence may be interrupted by loss of power, which is an pecially important consideration in some locations; or by some mergency situation where an operator may interrupt the auto- atic cycle as, for example, when operating an emergency stop utton upon the breakage of a tool.

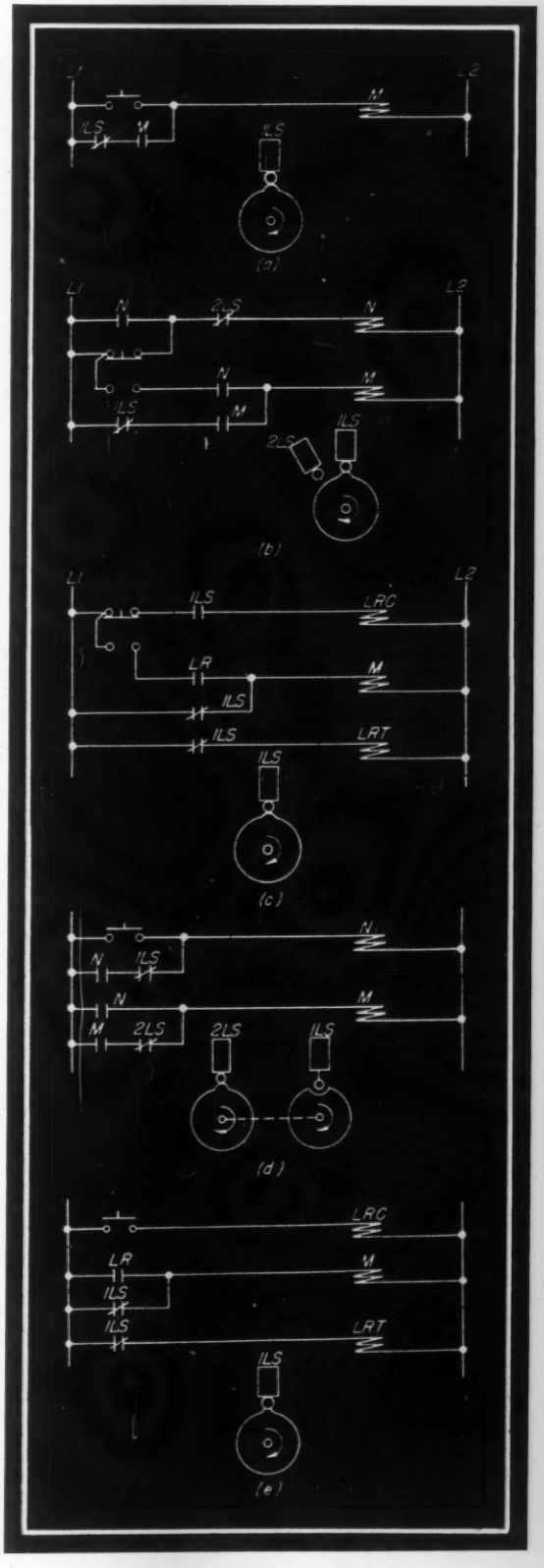
It is necessary in situation 1 to keep the control setup exactly e same after restoration of power, or after operation of the re- ming impulse, as before the interruption occurred. This some- mes is taken care of by the position of the machine and the arious devices. Sometimes it must be accomplished by mechan- ally-held electrical control devices as a part of the controller. he use of electrically-maintained circuits should be watched es- ecially closely, as the loss of a circuit by this means is likely to esult in the loss of the sequence which can then only be restored y manually operating an appropriate limit switch or manually losing the particular contactor or relay.

In situation 2, each motion can be operated manually and ndependently, as well as in the automatic sequence. This type f control is usually advantageous on machine tools, since it per- nits operating the independent motions separately for setup pur- oses. Care should be taken in the design of circuits so that no portion of the automatic sequence is put in motion while operat- ng manually. On some machines it is advantageous to arrange the circuits so that the automatic sequence cannot be initiated except when the various motions are all in the normal starting position.

The arrangement in situation 3 is seldom used, due to the hazard of equipment starting without notice on resumption of power and returning to the initial position.

Basis of all sequence-control arrangements is the single or nit cycle, or the movement of a single motion away from its riginal position and again back to this same position. A simple diagram for a circuit of this type is shown in *Fig.* 1*a*. Both electrical and mechanical arrangements are indicated. In the starting position, limit switch 1*LS* is held open by the cam. To start, the push-

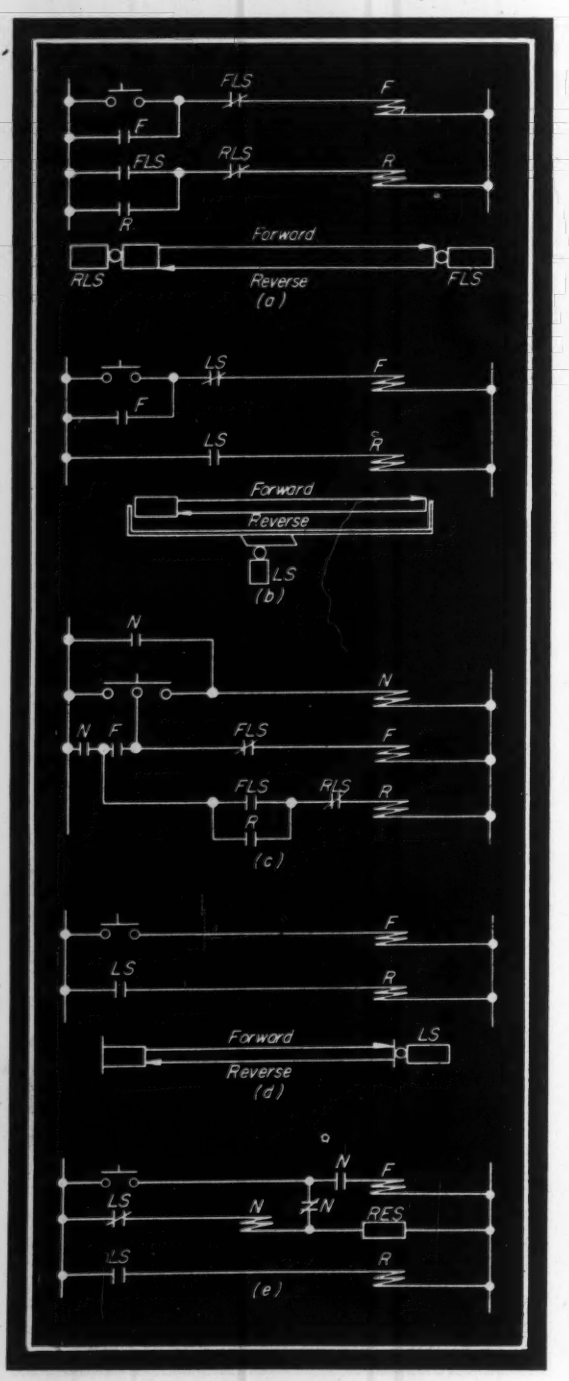
Fig. 1—Below—Single-cycle schemes for non-reversing crank motions



button is operated closing contactor *M*, causing the cam to rotate. As it begins to rotate the limit closes, and a maintaining circuit is established for the contactor coil. When the machine returns to its original position 1*LS* opens, de-energizing *M* and stopping the machine.

The next step might be as follows: Supposing that the machine should make only one cycle even though the pushbutton be held closed. Obviously, if the button in *Fig.* 1*a* were held closed, the machine would run continuously. The desired result might be accomplished as in *Fig.* 1*b* by utilizing an additional limit switch 2*LS* and an additional relay *N*. When the machine is in the starting position and the pushbutton is released, relay *N* will be energized through the normally-closed back contacts of the pushbutton. A maintaining circuit also exists around the button contacts. When the button is operated the contact of relay *N* completes the circuit between the pushbutton and contactor *M* and the operation of *M* is the same as in *Fig.* 1*a*. However, if the button is held down, when the cam comes around to limit switch 2*LS* this switch will de-energize *N* which will open, opening the pickup circuit to *M*, so that *M* will open when 1*LS* opens. Only by releasing the pushbutton is the pickup circuit established to *N*. Therefore, it is necessary to release and again depress the button before another cycle can be started.

Fig. 2 — Below — Single-cycle schemes for reversing motions. These are also adaptable to reversing electrohydraulic systems



System Has One Dead Spot

This scheme is satisfactory as long as the machine does not accidentally stop with the cam engaging 2*LS*. In this case there would be no pickup circuit to either *M* or *N*, and it would be impossible to resume operation unless some other method were employed, as for example a separate pushbutton for manual operation.

Another single-cycle arrangement is shown in *Fig.* 1*c*. It is particularly adaptable to intermediate cycles in a complete sequence, where the setup must be kept and the cycle must resume regardless of where it may have been interrupted. The initiating device is shown as a pushbutton and is assumed to remain closed all through the cycle. In actual application it might be a limit switch operated by the preceding motion of the sequence. In this arrangement a mechanically-held relay *LR* is used. This relay has two coils *LRC* and *LRT*. The former serves to close the contacts and the latter to open them.

In the starting position the limit switch is engaged so that the contacts are the reverse of the manner shown in *Fig.* 1*c*. With the pilot device released, a circuit is made to coil *LRC*, closing contact *LR*. When the pilot device is operated, a circuit is made to contactor *M*. As the cam moves, the limit switch is released. The limit switch establishes a parallel circuit to *M* and also energizes coil *LRT* which opens contacts *LR*. Thus the only circuit path to coil *M* is through the limit switch contacts. When the cycle is completed the limit switches open *M*, stopping the cycle. It is then necessary to release the pilot device and reclose it before another cycle can take place.

Variations Arranged to Suit Application

There are of course many variations to these arrangements to suit practical conditions. For example, the scheme in *Fig.* 1 might be varied so that it would not be necessary to hold the initiating pilot device closed until the limit switch has been released. Such an arrangement would be advantageous where the motion is rather slow. A typical scheme is shown in *Fig.* 1*d*.

The scheme in *Fig.* 1*c* might be adapted to the use of a momentary-contact initiating device but to retain the function of resuming after an interruption of the cycle, as in *Fig.* 1*e*. Basic features and typical applications of these single-cycle schemes are summarized in the accompanying tabulation.

All of the basic arrangements so far discussed have been

rotary or crank motions. Reciprocating motions operated by reversible operators must also be considered in discussion of cyclic control. Such devices as tapping machines, milling table feeding, etc., would come under this classification. A basic circuit might be as shown in *Fig. 2a*, or a variation of this using a single snap-action limit switch operated by a shipper rod is shown in *Fig. 2b*. Both of these schemes are indefinite as to whether they would resume after an interruption depending on what part of the cycle the interruption should occur. As isolated cycles they could be made definite in not operating by the addition of a so-called undervoltage relay as shown in *Fig. 2c*. This arrangement corresponds to *Fig. 2a* except for the addition of the relay.

Cycle To Resume Automatically

Supposing that, instead of protecting, on interruption, against restarting, it is desirable to insure that the cycle resume where it left off. A control scheme might be used exactly the same as in *Fig. 2a*; however, the mechanical arrangement of the electrical contactors *F* and *R* would be such that the maintaining interlocks would be mechanically held in position until the other contactor had operated. The same might be accomplished by substituting contacts of a mechanically-held relay in place of the maintaining contacts of the contactors *F* and *R* and by putting the coils of the mechanically-held relay in parallel individually with the contactor operating coils.

Many electrically-controlled cyclic processes use hydraulic operators instead of electric motors as a source of power. Such systems usually involve an electric motor driving a pump to supply hydraulic pressure, and electrically-operated valves for controlling the flow of fluid to the hydraulic operators. Since the valves have some inherent characteristics contributing to the overall method of control, a separate discussion is warranted.

Control of a hydraulic piston, which is the usual hydraulic operator, is often by means of a double-acting solenoid valve. Such a valve has two solenoids opposing each other. If one of the solenoids is energized, the valve admits fluid pressure to one end of the piston and relieves the other end. If the other solenoid is energized the action is reversed. Some valves are arranged to spring center to an intermediate position where both ends of the hydraulic cylinder are closed. Electrical control arrangements are the same as for an electric motor and *Figs. 2a, b* and *c* would apply.

Another type of valve is the maintained-position or non-self-centering type. This valve introduces characteristics in the overall system that are the equivalent of a mechanically-held electrical relay or contactor, and proper consideration must be given in the electrical circuit for these characteristics. The motion when once initiated continues its course until mechanically stopped, or electrically reversed. A sample circuit for such a system might be as shown in *Fig. 2d*. The scheme is adapted to a momentary-contact initiating device.

If the initiating device stays closed, some provision would be required to overcome the effect. *Fig. 2e* shows a form of circuit which might be adapted for this purpose. When prepared to operate, relay *N* will be closed, since the coil circuit will be energized through the resistor. Closing the pilot device energizes *F* through the contact of *N*, and *F* stays energized until the limit switch is reached. The limit switch opens *N*, which short-circuits its own coil through the pilot device and its own normally-closed contact. Thus as long as the pilot device is closed *F* cannot be re-energized even though the limit switch again closes. Opening the pilot device breaks up the short circuit and allows *N* to close, making ready for the next operation. *LS* also energizes the return contactor.

There are of course many variations on the basic idea of the single or unit cycle in applying it to actual situations, depending on the surrounding circumstances. The

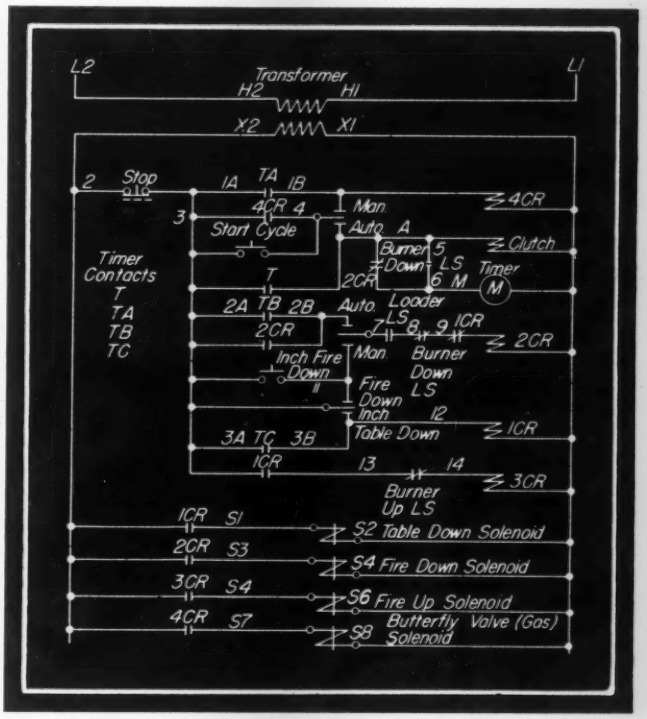
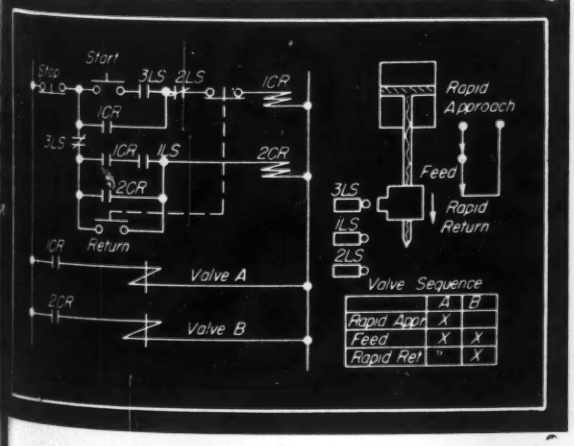


Fig. 4—Control diagram for a flame-hardening machine

Fig. 3—Below—Cyclic circuit with voltage-failure protection for hydraulic-feed control



initiating device employed; whether or not the cycle is continuous; whether it may be interrupted to allow an intermediate event to take place; the protective interlocking required with other events; and the manner in setting off the succeeding events—all have an effect on the detail arrangement of the control scheme.

Fundamental characteristics of controls for reciprocating motions are also listed in the accompanying table.

In most circuits involving cyclic control it is desirable to prevent automatic restarting after interruption as, for example, by failure of power. This is true whether the process is to resume from where it was discontinued or whether the motions are to be returned to an initial starting position. In some instances this requires an electrically-maintained control relay with no other function. Such a relay would be operated by pushbuttons marked, for instance, reset or cycle start and stop. This is particularly true of cycles that are to resume.

Protecting Hydraulic Circuits

Often situations arise in the control of electro-hydraulic systems where combinations of valves produce a certain result. With such cycles it is possible to set up a control circuit which has undervoltage and stopping protection at all times, even during the actuation of any limit switch, without the use of separate undervoltage relay. *Fig.* 3 illustrates a simple drilling machine with hydraulic feed control utilizing such a method.

Arrangements like this require that there never be a transfer in which all devices open. In the illustration the sequence of operation is such that first 1*CR* closes, then 2*CR* closes, then 1*CR* opens, then 2*CR* opens. This circuit contains the requirement that, in case the cycle is interrupted, it is necessary to return to the starting position before it can be resumed. The valves in this case are not of the double-acting type but are single valves, which reset when power is removed from the solenoid coil.

In other hydraulic circuits the undervoltage and stop feature may be obtained in the control of the hydraulic pump. This might be accomplished with the schemes shown in *Figs.* 2*d* or 2*e*. With hydraulic systems involving nonself-centering valves, control of the pump or control of an additional master valve are the only methods by which low-voltage protection may be obtained.

Interlocking Several Motions

Some machines and processes involve several motions or unit cycles, and therefore require a control scheme to combine and interlock the various unit cycles adequately. The manner of combining circuits is as endless as the number of processes or machines which can be devised. Perhaps, however, an examination of a representative circuit will bring out at least a few possibilities.

In *Fig.* 4 is illustrated a control for a flame hardener. It involves two motions and the operation of a valve, with part of the sequence under the control of a sequence time control and part controlled by the position of the machine. This device consists of a table which raises and lowers into a quenching bath under control of a table-down solenoid valve. Release of the solenoid raises the table. Above the table is a burner which raises and lowers by means of the so-marked solenoid valves. Loading of the table is manual. The automatic cycle has the following sequence of operation:

1. Cycle start button is operated
2. Timer starts
3. Contacts *TA* close immediately to open gas valve
4. After twenty seconds contacts *TB* close and burner starts down if loader position is correct; timer is stopped while burner moves down
5. With burner down, timer resumes for 30-second heating period
6. Contact *TA* opens and gas valve is closed
7. At the end of twenty seconds contact *TC* closes, dropping table into quenching bath for additional twenty seconds and also raising burner
8. After seven seconds timer contact *T* opens, resetting entire cycle.

This controller, besides having a complete automatic cycle, can be operated manually through the selectors and pushbuttons. The timer master controller is of the automatic reset type, resetting upon de-energizing its clutch coil.

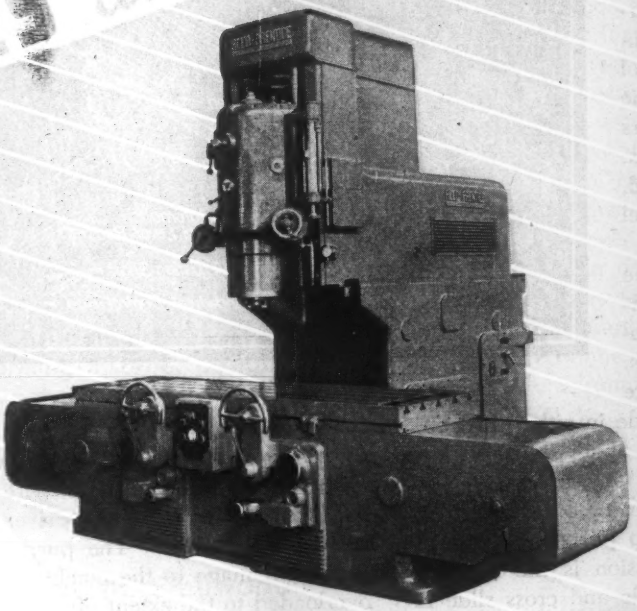
Has Three Hydraulic Motions

In the head illustration is shown a Davis & Thompson Co. milling machine. In approximately twenty steps this machine completes all of the rough or finish milling of the channel sections on connecting rods for aircraft engines without the use of complicated and elaborate follow-up or tracer systems. The machine has three hydraulically operated feed motions consisting of a rotary motion of the table, a straight line motion of the slide on which the table is mounted, and a horizontal motion of the cutting head at right angles to the slide motion.

Since the machine has a definite sequence of motions all interrelated to one another, the cycle is set up by a sequence of limit-switch operations. Interesting features of the control operation are that the motions must all be in an initial starting position before the cycle start button can be operated. To get all of the motions into their respective starting positions, it is necessary only to operate a single return pushbutton. This button is especially useful to reset the machine after cycle interruptions due to power failures and emergency stoppages. Complete control of the automatic cycle is obtained with only ten relays in addition to the several motor contactors including spindle, hydraulic pump and coolant pump motors.

While an examination of the control scheme for such a machine is beyond the scope of this article, it can be said that it incorporates many of the fundamental considerations that have been covered. The purpose of the article has been to emphasize such considerations and to show that there is always a systematic approach to problems involving the control of sequence or cyclic processes.

DRIVES and CONTROLS



Hydraulic Transmission Facilitates Feed Control

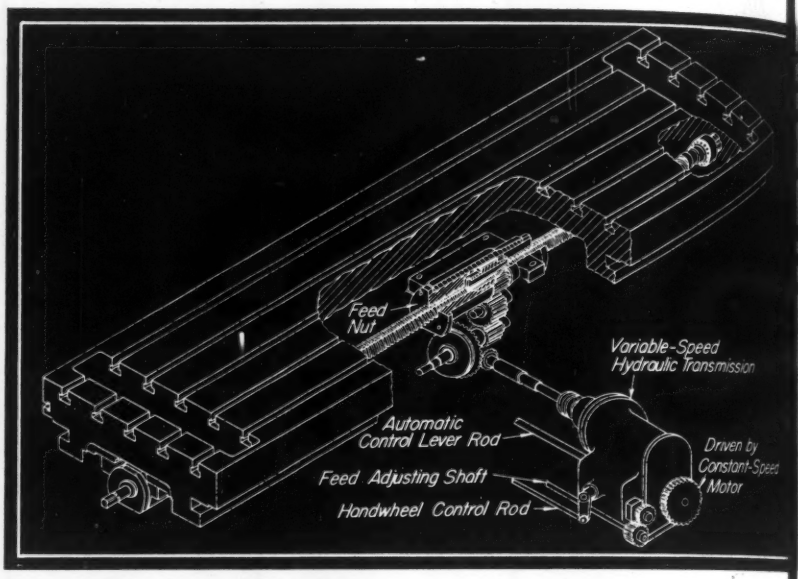
By R. L. Rougemont
Chief Engineer
Reed-Prentice Corp.

TWO hydraulic transmissions, identical in their characteristics, are utilized for the feed drives of the table and the cross slide on the vertical milling machine shown above. They provide a stepless, variable-feed drive at constant torque together with one fixed rapid-traverse speed in both directions. The variable feed drive has a speed ratio of approximately 50:1, giving a feed rate of ½ to 25 inches per minute. The rapid-traverse speed has a fixed rate of speed of 180 inches per minute. Built as a single unit, the hydraulic transmission is a variable displacement multipiston pump which delivers oil to a fluid motor of the same type.

Both transmissions are driven through gears by only one 5-horsepower electric motor at a constant speed of 1420 revolutions per minute. The variable speed shaft of the transmission is the hydraulic motor or output shaft and its speed is varied by changing the displacement of the pump and hydraulic motor. Control of speed is effected by solenoid-operated hydraulic valves mounted on the hydraulic unit itself.

For selecting feed or rapid traverse in either direction a five-position sliding valve is also built into the transmission unit and operated from a lever which is located at the front of the machine.

Feed rate is determined by the position of an adjusting screw or an airplane-type handwheel, also located in front of the machine. Both control the displacement of the hydraulic pump and motor through wobblers carried by the shafts of these units. The adjusting screw is used for fixed, preselected feed rates, the handwheel for variable feed rates.



Control of variable motion for the table and cross slide with these handwheels is one of the advantages derived from hydraulic transmissions. The controls are effortless to operate and will move the table or cross slide in either direction. Maximum motion of handwheels is about 90 degrees in both directions, neutral point being in the vertical position. Both table and cross slide can be moved in increments of .001-inch for setting up purposes. Another advantage of this type of transmission is the fact that screws are used for driving the table and cross slide instead of a hydraulic cylinder, as shown in the accompanying diagram of the table drive.

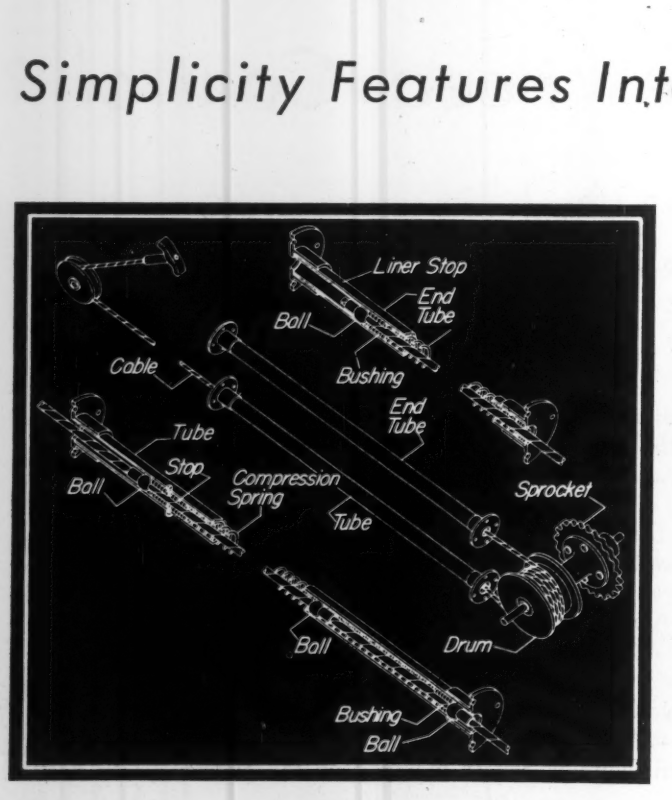
Besides controlling the five-position sliding valve and variable pump, the control levers operate a set of limit or auxiliary switches which in turn energize or de-energize the solenoid valve and rapid-traverse clutch solenoid. The function of these levers is to establish which method of control is effective: Automatic with preselected-feed speed rate or the handwheel with variable feed rate.

Spindle motor and motor-driven hydraulic units are electrically interlocked. The purpose of this interlock is to prevent damage to the machine in the event the cutter is overloaded to the extent of stalling or near-stalling during a cut. If the table continued to feed when the cutter is stalled, damage to the cutter and spindle head would result. Therefore provision has been made to stop the table or cross slide automatically whenever the spindle motor is called upon to deliver approximately 300 per cent of full-load rating.

Simplicity Features Intermittent Manual Drive

By C. W. Sweetser

Assistant Chief of Structures
Grumman Aircraft Engineering Corp.



NOW being successfully used in a new military airplane to solve a particular drive problem is the simple manual device shown in the accompanying drawing. The problem was to move manually an installation weighing about 30 pounds a distance of four feet, positively with a cockpit control whose available motion was limited by space considerations to about eight inches.

The principle of this device is one used constantly in hoisting and rigging work, in which several turns of a rope are wound around a smooth rotating drum and power is applied to the rope through friction maintained by keeping the "loose" end of the rope taut. In this device the cable drives the drum and the loose end is held taut by a spring.

The drive operates as follows: A cable runs from the actuating handle around a pulley and through a fixed tube

which a compression spring expands against a fixed p and a sliding bushing, the latter bearing against a l swaged to the cable. Another ball, also swaged to cable, limits the cable travel when it contacts the stop the tube. The cable extends from this tube in several s around a drum which in turn is attached to a sprocket. roller chain engaging this sprocket transmits the result- motion to any point required. The cable continues the drum into an end tube which is mounted above d in the same plane with the other tube. In the end tube, a bushing bearing against a swaged ball is forced by another compression spring against its stop, which is a liner in the tube. A roller and guide not shown in the illustration are provided around the drum to prevent improper threading or overlapping of the cable.

When the cockpit handle is pulled, the cable grips and rotates the drum, as the other end of the cable is restrained by its spring. The other spring has also been compressed to provide energy for the return of the cable to its original position. The length of cable between the end balls in each tube has been so determined that, on the return stroke, the coils around the drum immediately expand until they are free of the surface of the drum. This allows the system to "free wheel" between strokes.

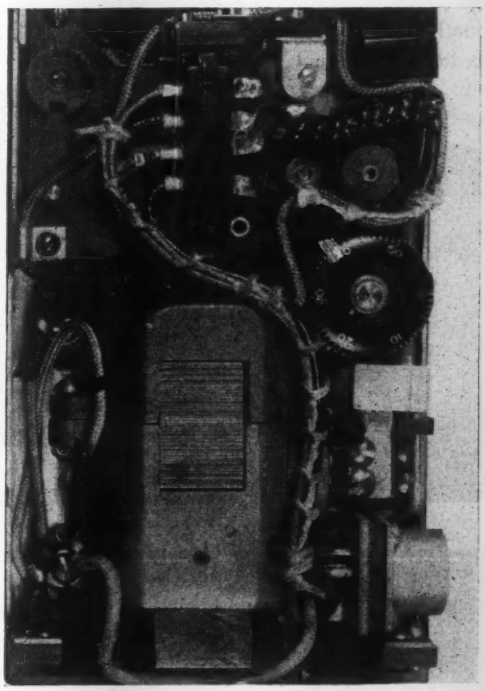
With the unit arranged as described a pumping motion at the handle effects an interrupted but positive rotation of the sprocket in one direction, which can continue as long as is necessary to give the required chain motion.

The extreme simplicity of the device, and its resulting reliability, recommend it particularly for use in combat aircraft, in which maintenance is always critical.

DESIGN being universal, engineering developments in one field have useful application in others. With this thought in mind it is hoped that the engineers' symposium on these pages, covering drive and control problems in specific machines, may suggest the answers to questions arising in other designs

Dependability Achieved With Heaters

By Frank W. Stellwagen
Engineer-in-charge
Fairchild Camera & Instrument Corp.



ABOUT 1934 our first heating problem was encountered on a motor-driven timing device, called the intervalometer, for automatic tripping of n aerial camera at predetermined intervals. The intervalometer failed epeatedly during high-altitude flight because of the low temperature. Due to these failures, about 400 units had to be rebuilt with minimum hanges to insure reliable operation under high-altitude flight conditions. A two-watt, wire-woumd resistor and a thermostat, which closed the eater circuit when the temperature was below 45 degrees Fahr. and pened when the temperature was above 90 degrees Fahr., were installed in the mechanism. Insulation lining was placed in the inner alls of the case to reduce heat losses.

Experimental tests of this thermostatically-controlled heater resulted n holding the torque demand of the mechanism essentially constant from lus 90 degrees to minus 40 degrees Fahr. Use of thermostatically ontrolled heaters and thermal contacts in motors results in a smaller otor size, lower current demand, and a more uniform mechanism perormance over the extended temperature and voltage range required for erial photography.

Another advantage of the heater was that moisture particles, condensed on the ball bearing surfaces of the drive motor and echanism, did not freeze. In the past these frozen moisure particles had jammed the mechanism so tightly that even an oversized motor would not drive the mechanism nd would cause frequent breakage of gears and pins in he mechanism.

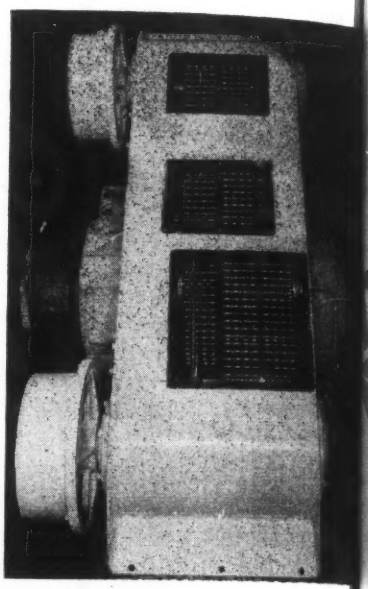
Success of the intervalometer redesign sold us on thermoatically-heated mechanisms. More recent applications, s in the Fairchild gun camera, have been taken care of in ss space and with more uniform heat distribution by using a smaller thermal protector and a flexible wire-wound resistor covered with fiberglass sleeving.

The flexible resistor permits winding the heater element throughout the mechanism. This allows the most critical locations to be uniformly heated. It is preferable to having all the heat concentrated in a rigid resistor depending on air conduction to circulate heat through the enclosed area. This is particularly useful where a film magazine

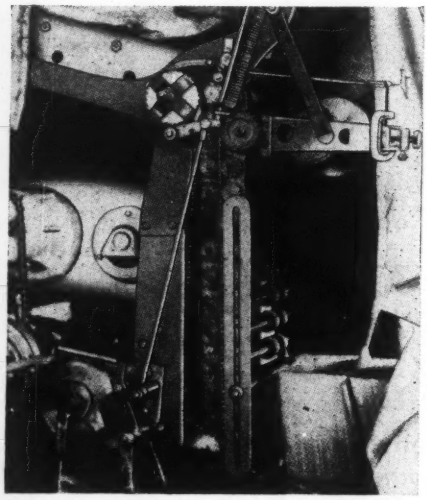
(*Concluded on Page* 170)

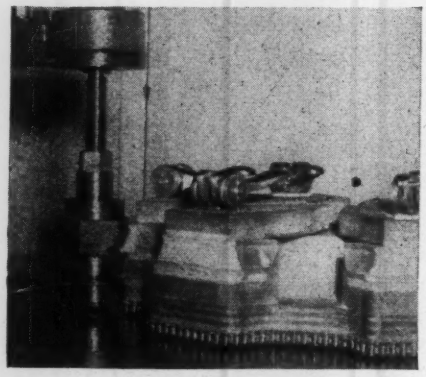
Novel Applications of Roller Chain

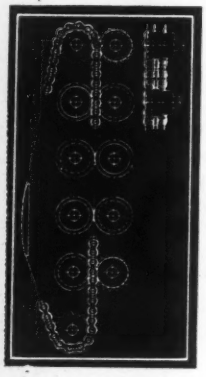
DIFFICULT design problems often can be solved by application of standard parts or units in a manner not generally considered conventional. The applications discussed on this page show how three tough problems were solved by using standard roller chains.



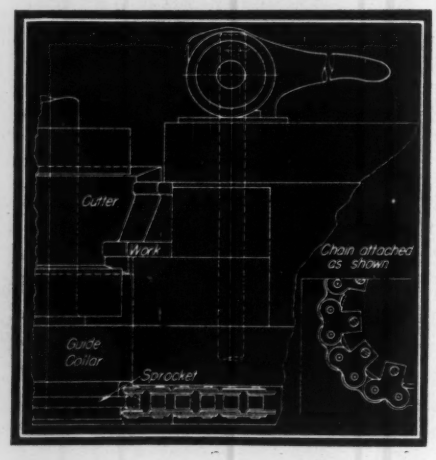
IN the shaper (shown below) for forming wood parts, Baldwin-Rex roller chain is fitted around the bottom edge of the form holding the work to be shaped. The form is revolved by a feed sprocket engaging the chain and is held in contact with the guide collar by air pressure.



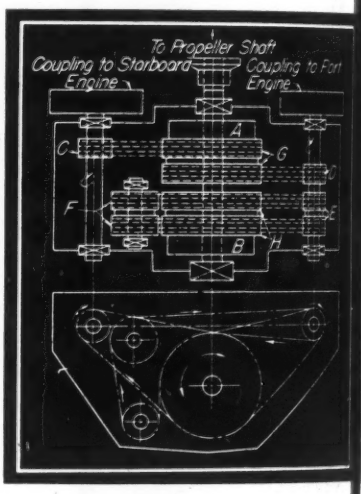




IN the printing press mechanism illustrated above, two rows of shafts carrying rubber friction rolls must rotate in opposite directions to feed paper into the press. On one set of sprockets the outside row of teeth is machined to the root diameter, and on the other set the inside row is machined. With one row of sprockets driven from the outside strand of the Baldwin-Rex double chain and the other row of sprockets driven from the inside strand, there is in effect produced a rack and pinion drive with an endless rack driving the chain pinions.



THE marine transmission (upper right) makes possible a reversing propeller-shaft drive using two non-reversing engines with the combined power of both engines effective in either direction. Diamond multiple-roller chain is threaded over the sprockets as shown. With both engines running in the same direction the propeller shaft will turn clockwise when clutch *A* is engaged and clutch *B* disengaged. Sprockets *C* and *D* then drive the propeller shaft and sprocket *H* floats free on the shaft. To reverse, clutch *A* is disengaged and clutch *B* is engaged. Sprocket *G* floats on the propeller shaft, permitting power to flow from sprocket *C* to *D* as well as from sprocket *E* to *H* in conjunction with *F*, the idler sprocket which is required to effect reversal. Action is smooth, and "full astern" to "full ahead" and vice versa is achieved in only a few seconds.



SERVOMECHANISMS—How They Work

By R. H. Rogers
Industrial Engineering Division
General Electric Company

BACK in the days when slaves were used to provide motive power for sailing vessels, some ships were equipped with a mechanism which permitted the skipper to steer the vessel with the exertion of but little energy. His tiller was fastened to a vertical shaft at the bottom end of which, below decks, was attached a long horizontal pointer. This pointer was positioned directly above the power tiller, movement of which turned the ship's rudder. A number of slaves were used to move the power tiller, taking their direction from the pointer over their heads.

Actually, while the masters of these ancient ships no doubt referred to the device by some other name, what they had was a servomechanism. Although it was extremely crude and its profligate waste of manpower would not be tolerated today, nevertheless it did substantially what modern servomechanisms do. When the skipper moved his signaling tiller, it took time for the slaves' reactions to take effect. This constituted "lag" in the system. Again, after the slaves had moved the power tiller they could not stop it with the ease enjoyed by the skipper with his signal tiller, so they overshot the mark

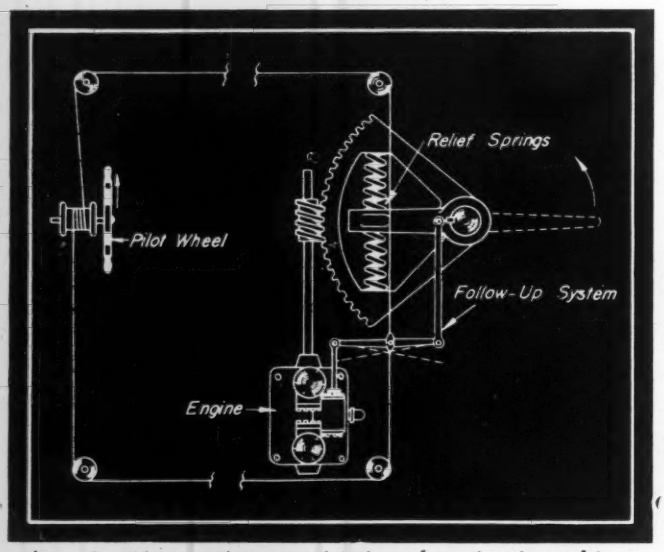
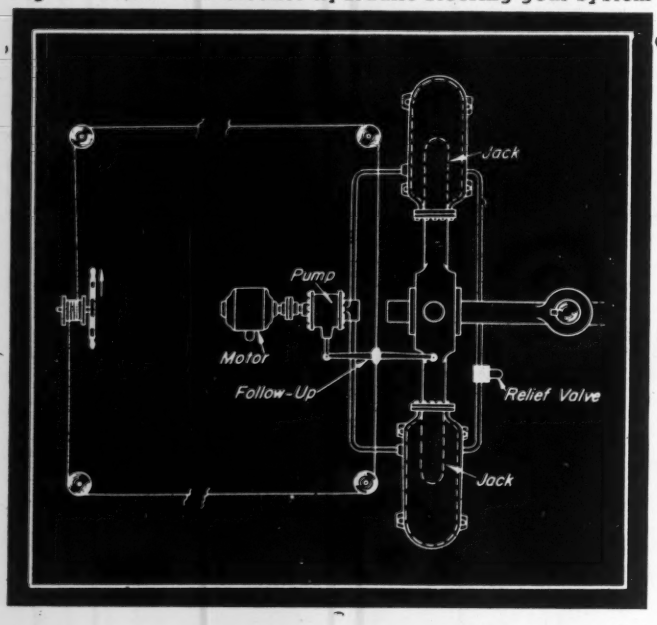


Fig. 1—Above—Servomechanism for steering ships

Fig. 2—Below—An electric-hydraulic steering gear system



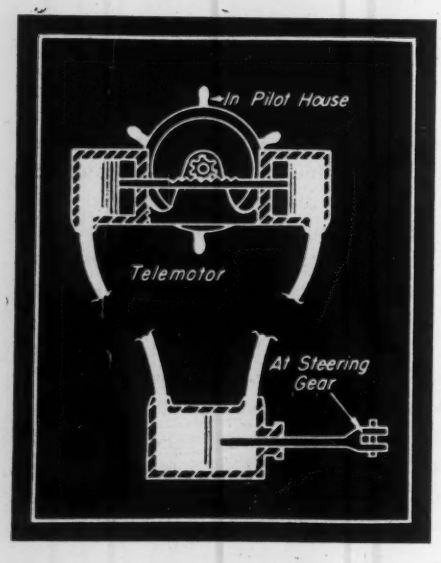


Fig. 3—Left—Remote control by hydraulic "telemotor"

Fig. 4—Below—Differentialscrew type of follow-up

and had to pull back to match the pointer setting. Fina
extremely small movements of the pointer could not be r
ognized as signals by the slaves, with the result that t
system had a "blind spot" area.

These three faults; lag, overshooting and blind spot, p
sist in all servomechanisms to this day. Although th
have been reduced tremendously, they are always pres
nevertheless.

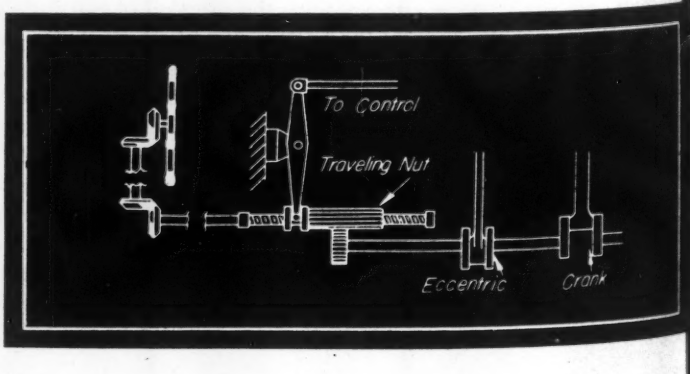
In broad terms, servomechanisms may be likened
long, strong arms. They extend the operator's reach, s
ply power to do the work, and move the load practica
in unison with the operator's control, starting and stopp
when he does.

That last mentioned, self-stopping feature is essen
to the definition. To make this clear, consider a mo
steering gear. When the pilot gives the wheel one spo
to port, the rudder goes to port a proportionate amo
and stops there. By contrast, an ordinary electric rem
control starts a motor when the control is turned on
the motor continues to run until the control is turned
Such a system on steering gear would be unsatisfactory
it would put the rudder hard over with every slight t
of the pilot wheel.

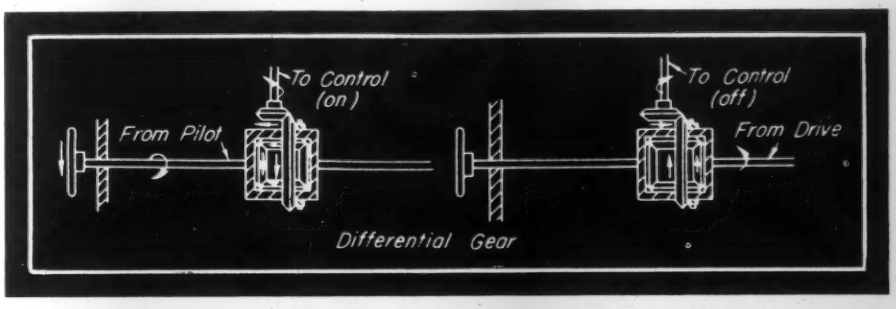
Governing, regulating, leveling and positioning a
among the functions of automatic servomechanisms and
such systems must comprise four essential features:

1. A signal device through which the order to be execut
is passed. It may be a rope, an oil line, a selsyn,
potentiometer, a bridge circuit, a phototube, etc.
2. A determining member which decides which way acti
is to take place and the urgency of the call. It may
a reversing throttle, a four-way hydraulic valve, a co
plex air controller, or an extensive electric control syste
3. Motor power to do the work. This may be an engine,
hydraulic cylinder, stored air, an oil or electric motor
which some system of gearing may be connected.
4. A follow-up, feedback, or restoring means which sto
action when the order has been executed. This may
a simple lever, a valve, a differential gear, or an electr
system closely related to the signal system.

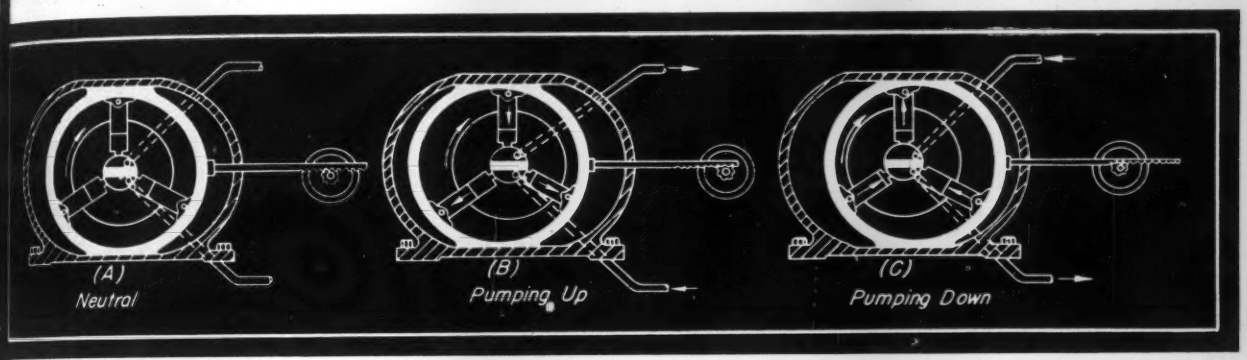
An ideal servomechanism would maintain the respons
in exact correspondence with the signal at every instant
time. This is impossible of attainment since the differenc
between the signal and the response is the very thing th
makes the system work. This lag in response after signa
may be quite broad in some applications without doin
any harm while in others it may be reduced to as ne
zero as possible. In modern systems that is near indee



5—Right—Closed-system type differential gearing often is d to provide restoring action



6—Below—Radial-piston type, ersible, adjustable-flow pump ch as is employed frequently in hydraulic servomechanisms



Another characteristic hich may or may not be itical according to applica- n is blind spot. Even at st, the signal must move a ile before action can be rted and the blind spot is ice the width of that little ovement. The refinement d sensitivity of vacuum be circuits have worked onders in narrowing the ind spot.

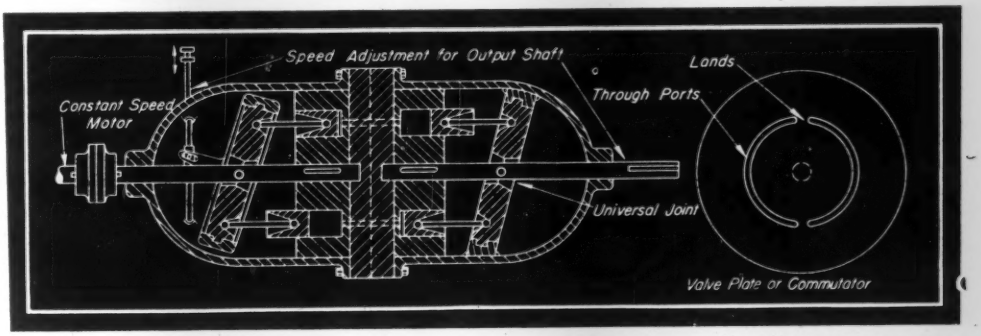
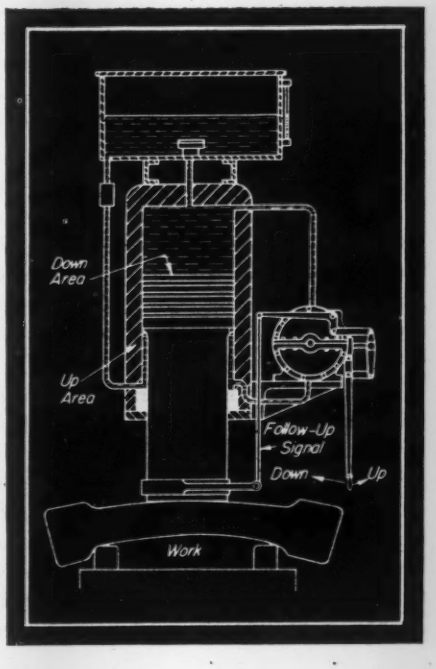


Fig. 7—Above—Axial pistons are employed in this adjustable-flow pump

Fig. 8—Below—Press control



The third characteristic is overshooting or hunting, in which the response ns on ahead after the signal has stopped, then comes back and perhaps avers a time or two before stopping. Heavy inertia loads aggravate this con- tion. Electrical systems provide ready means for offsetting the tendency. ntihunt circuits, limit switches, and dynamic braking are effective.

The illustrations of this presentation show only essential features and will id in the explanation of the general principles of most types of servomechan- ms in active use.

Fig. 1: One of the earliest and still a common form of servomechanism s the steering gear for ships. Signal is transmitted to the steam engine throttle y wire cable, shafts and gears, or by hydraulic means. The "differential alve" on the engine controls speeds and reversals by simple push-pull action n a piston valve. The follow-up or restoring function is accomplished by a ever which fulcrums first about the connecting rod from the steering gear to pen the throttle, then about the signal connection to restore the piston to eutral.

This is an example of where wide blind spot is a virtue as it cuts down uddler movement from about twelve per minute to perhaps three per minute y elimination of the small angle signals sent down by a nervous quartermaster.

Fig. 2: Electric-hydraulic steering gear has several advantages over the ciprocating steam engine. An electric motor drives an adjustable, reversible- ow oil pump. Again a push-pull member provides control. When action is quired, oil is pumped from one jack around into the other, forcing the tiller

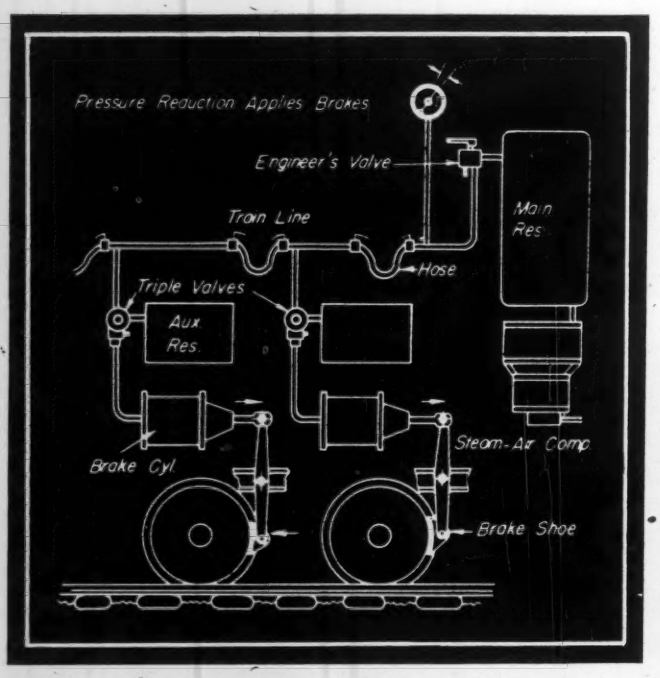


Fig. 9—Pneumatic servomechanism on railroad air brakes

over as required. The follow-up is similar to that in *Fig.* 1. Stored energy in the motor rotor and pump cylinder block is not transmitted to the load, hence there is little tendency to overshoot. The motor and pump run continuously, so there is only slight delay in getting action. The hydraulic jacks provide a great speed reduction between motor and rudder stock with few wearing parts, all well lubricated. Heavy shock loads on the rudder are relieved by by-pass valves between the jacks, thus doing away with mechanical means for relief.

Fig. 3: A superior means for transmitting signals is provided by the hydraulic "telemotor", an arrangement of cylinders and oil pipes. This system operates well for distances sometimes well over 600 feet, where cables or shafting would impose a considerable friction load and much lost motion.

Fig. 4: This diagram shows a differential screw type of follow-up which often is used with *Fig.* 1. The pilot wheel, by shafting and gearing, turns a screw to move a nut to which the push-pull member is attached. The engine, through suitable gearing, screws the nut back to restore the control to "off".

Fig. 5: Differential gearing often is used with servomechanisms to provide restoring action in a so-called "closed system." Turning the pilot shaft turns the control shaft while the restoring shaft stands still or is following. When the pilot shaft stops, the restoring shaft continues until the control shaft is in the "off" position. There are many ingenious versions of differential gearing as applied to the follow-up function of mechanical and hydraulic servomechanisms.

Fig. 6: Hydraulic servomechanisms for machines, steering gear, turrets and other heavy-duty slow-moving jobs are served by reversible, adjustable flow pumps. One radial-piston type is shown. The cylinder block carrying seven or more pistons is driven at a constant speed by a motor. The stationary bridge across the center port divides it into suction and discharge sides. Pistons are guided by a movable ring (white in the diagram) against which the piston shoes are pressed by centrifugal force. As shown in (*A*) the ring is on center and the pistons do not move relative to the cylinder block. In (*B*) the lower [illegible] tons are moving outward establishing suction and [illegible] upper ones are moving in, thus providing pressure. [illegible] centricity of as little as 1/64-inch will start positive [illegible] ing action. In (*C*) the push-pull member has been [illegible] out and oil is flowing in at the upper pipe and out at [illegible] lower one. One-inch diameter, one-inch maximum [illegible] pistons are usual so a delivery of 6000 cubic inches [illegible] minute is possible at 1200 revolutions per minute. Appl[illegible] to a 12-inch jack, this produces a movement of [illegible] 50 inches per minute with thrust amplification of [illegible]

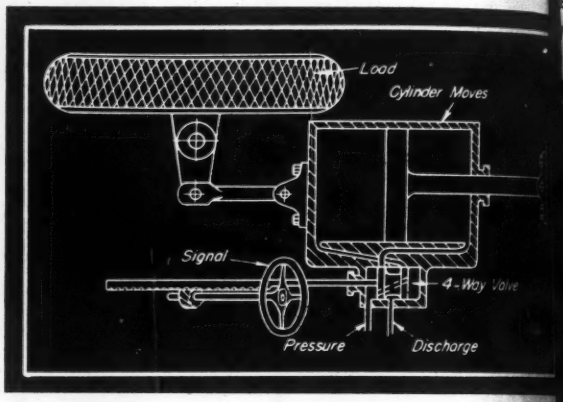


Fig. 10 — Above — A simple hydraulic [illegible] mechanism with rack-controlled 4-way valve

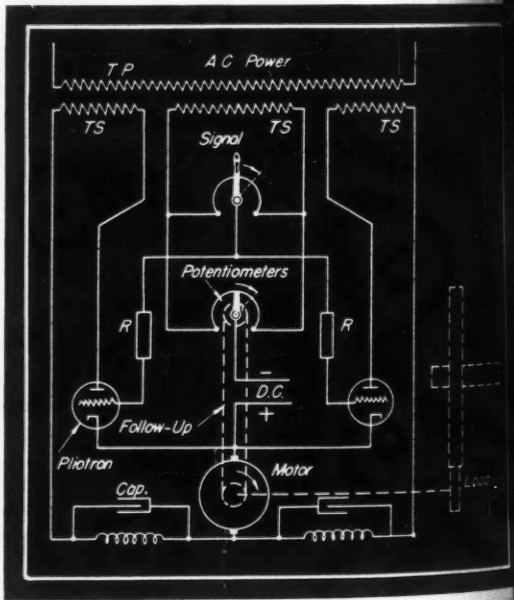


Fig. 11—Above—All-electric servomechanism [illegible] trols small a-c, reversing, split-field motors

Fig. 12—Below—Direct-connected selsyn [illegible]

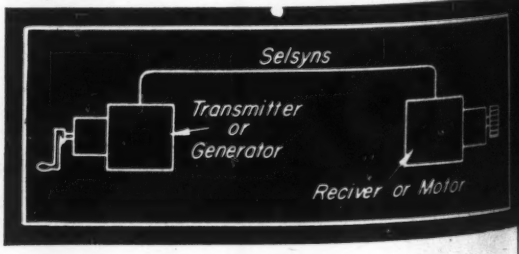


Fig. 7: Another type of reversible, adjustable-flow pump uses axial pistons. Again the drive is constant speed. When the guide at the left is vertical the pistons do not move relative to the cylinder blocks. Inclining the guide more or less either way determines the flow of oil in quantity and direction. The motor half at the right, which may be at a remote point connected by pipes to the pump part, has a fixed guide. Speed of rotation of the oil-motor output shaft is proportional to the slant of the guide in the pump member. There is a striking analogy here to electric systems. The pump furnishes constant frequency alternating currents which are commutated by the stationary valve plate to direct currents of oil. When a cylinder

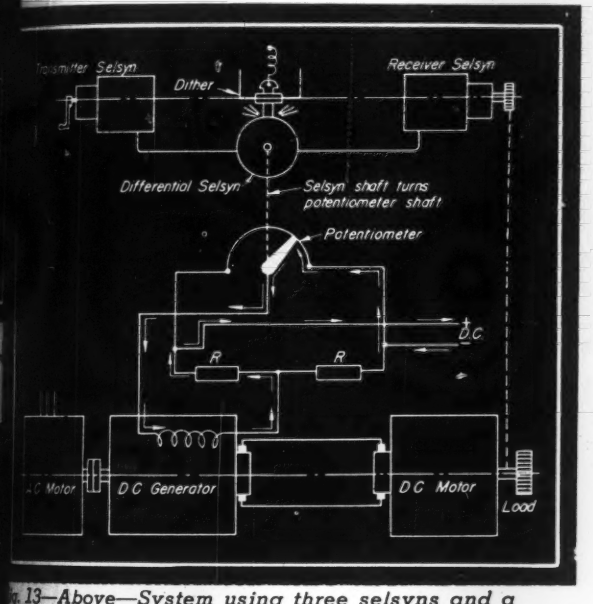


Fig. 13—Above—System using three selsyns and a bridge circuit controls field of d-c generator

Fig. 14—Below—Geared selsyns reduce "blind spot"

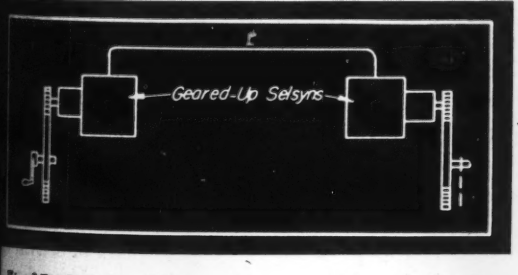
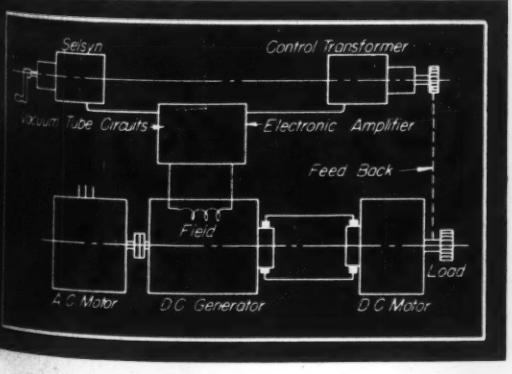


Fig. 15—Below—This system cuts down on "lag"



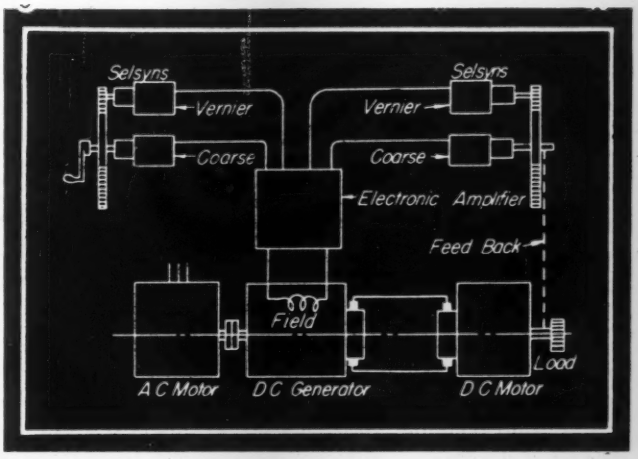


Fig. 16—Above—Geared selsyns keep system "in phase"

port is passing the "land" between the valve ports the piston is neither going in nor out. On the other or motor end, the direct currents of oil are inverted and have adjustable frequency proportional to the flow from the pump. It will be seen that this frequency determines the speed of the oil motor.

Fig. 8: A pump of one of the types just described may be used on a machine such as the straightening press shown. The pump is driven by a constant-speed motor and is connected to a hydraulic cylinder. In the down direction, because of the great area of the piston, the speed is slow and the thrust is powerful. On the up stroke, the piston area is relatively small and up travel is fast but with little thrust. The operator's lever connects with the push-pull member which also is connected with a follow-up system. The operator has fine control over the high-pressure thrust.

Fig. 9: The railroad air brake furnishes a good example of a pneumatic servomechanism and the one to which we are most often exposed. Under normal conditions, the main reservoir is charged and the train line and auxiliary reservoirs carry about 80 pounds pressure. The engineer discharges some air from the train line, reducing its pressure by perhaps seven pounds. The "triple valve" now passes a moderate amount of air from the auxiliary reservoir to the brake cylinder and brake-shoe pressure is applied proportionate to the train line reduction that was made by the engineer. The wave of depression travels back through the train line perhaps a mile or more in a few seconds, setting the brakes on each car in succession from locomotive to caboose. This ordered progress of braking prevents pulling the train in two. The engineer can now increase or decrease the braking effect by lowering or raising (note opposite effect) the train-line pressure. Restoring the train-line pressure releases the brakes and recharges the auxiliary reservoirs. Bursting of hose or train pulling apart sets all brakes on both sections in "emergency application", in other words, it fails safe. The follow-up system is in the triple valve, an aggregation of pistons within pistons, slide valves, tapered ports and check valves all comparable in ingenuity with many of our best electric designs.

Fig. 10: A utility servomechanism is typified in this steering gear for heavy vehicles such as portable cranes on landing fields. The general idea is applied to heavy loads to be lifted

and lowered at low or moderate speeds. In some applications the piston moves instead of the cylinder as is shown. The four-way valve takes many forms and it may be operated by a remote-control system. Oil for the cylinder is usually furnished by a one-way constant-speed pump. Surplus oil is returned to the pump through a discharge valve. Slide valves of any kind are conducive to liberal blind spot, because valve lap must be provided and leakage causes the motion to stop with the port slightly open. To reverse, the valve must be traveled some extra distance before a working opening is established.

Fig. 11: This all-electric servomechanism is suitable for small motors of the reversing split-field series alternating-current type. The motor circuit utilizes two pliotrons at the same potential and with a split anode transformer supplying the individual forward or reverse fields through the bi-phase half-wave rectifier circuit. Capacitors permit more current to flow from the half-wave sources. The resistance bridge contains two potentiometers, one moved by hand and the other driven to follow up as the motor

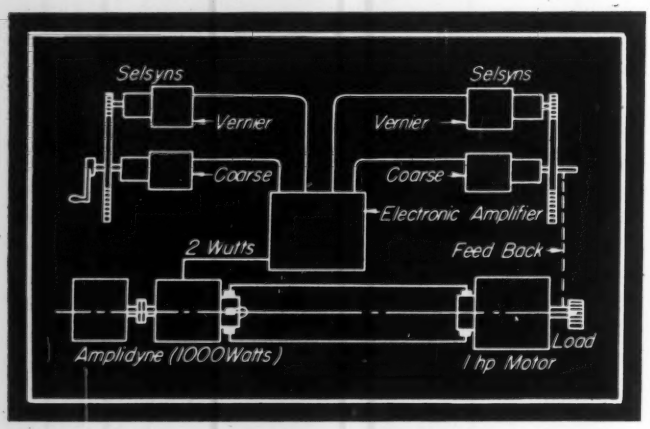


Fig. 17—Power step-up in an electronic servosystem

turns. When the two potentiometers are in practical correspondence at any point the motor stops rotating.

Fig. 12: The familiar transmitter-receiver selsyn system or generator-motor system will serve to introduce the line of refinements that have become so important in recent years. This generator-motor system is used most generally for indicating because such service does not impose much torque load on the motor member. At correspondence, torque is zero and it increases to a maximum at 90 electrical degrees out-of-phase. When rotation of generator and motor are maintained, one makes just as many revolutions per minute as the other but the motor lags according to the torque it is delivering. Depending upon size and type, dead swing one way may be from about three degrees to a little less than one degree.

Fig. 13: Selsyns, including a differential selsyn, can be arranged as a servomechanism by making use of a bridge circuit. When the transmitter selsyn is turned, the electrical differential established between it and the receiver selsyn energizes the shaft of the differential selsyn, causing it to turn in the proper direction an amount comparable to the rotation of the transmitter shaft. The potentiometer arm, driven by the differential selsyn shaft, throws out of balance the resistances in the bridge circ and, depending on which way the potentiometer a moves, current will flow from the direct-current sour through the field of the direct-current generator in one rection or the other. (Direction of current flow with potentiometer setting shown is indicated by arrows.)

Thus the generator feeds the direct-current motor w impulses which will turn it in the desired direction. A g train, or other transmission means, establishes the fe back from the motor output to the receiver selsyn a after the shaft of the receiver selsyn has turned throu the same number of degrees as was originally put into t transmitter selsyn, the electrical differential effected w drive the shaft of the differential selsyn in such directi as to turn the potentiometer arm back to center. T action balances the resistances of the bridge circuit a no current will flow through the direct-current generat field, with the result that the entire drive stops. "Dith (continuous vibration) is applied to reduce static bru and bearing friction in the differential selsyn to make more sensitive to small changes.

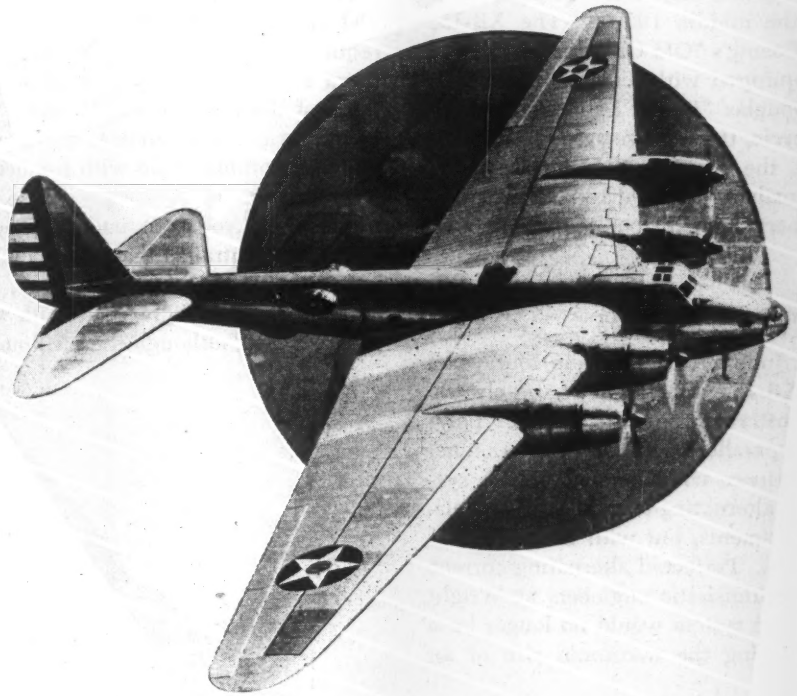
Fig. 14: In order to reduce the space in which acti is absent (blind spot) because of weak torque, the selsy can be geared up, for instance, 30:1, as compared wi director and load. By this means the dead range of s 120 minutes of arc becomes only four minutes at the wo shafts.

Fig. 15: Where selsyns are directly wired together, where an electrical differential is utilized for driving third selsyn as in *Fig.* 13, the amount of lag incurred the system is dependent on the sensitivity of the selsy themselves to the small energizing voltages involved a the torque required to handle whatever light load is co nected to the driving selsyn.

The diagram of *Fig.* 15 shows how lag can be cut dow considerably by using much smaller voltage differenti The microscopic voltage set up between the selsyns is ele tronically amplified and the power output of the amplifi is, in this case, used to energize the field of a direct-curre generator. The unit referred to in the diagram as a "co trol transformer" is merely a selsyn with a high impedanc

Fig. 16: If the director (transmitting selsyn) of *Fig.* 1 is turned too far or too fast, the two selsyns may get o of phase. This is especially likely to happen with iner loads and with geared units as in *Fig.* 14. A means circumventing this possibility is to use pairs of selsyns f transmitting and receiving, each pair geared together shown in *Fig.* 16. The selsyns carrying the large gea are referred to as "coarse", and those fitted with the sma gears as "vernier". The coarse selsyns serve to hold t system in phase while at the finish the vernier units co in to mop up the error. The rest of the system is the sam as that employed in *Fig.* 15.

Fig. 17: Using the selsyn pairs and electronic amplifi of *Fig.* 16, the fields of an amplidyne unit can be ene gized from the vacuum tube circuit. In this case the tw watt output of the electronic amplifier is further amplifi to 1000 watts by the amplidyne, the output of whi energizes a one-horsepower motor. Geared feed-back i the same as in previous diagrams.



Will Aircraft Switch to 400-Cycle Auxiliaries?

By J. D. MINER, Jr.
Manager, Design Engineering Department
Westinghouse Electric & Mfg. Co., Lima, O.

IT IS slightly over two years since the Army Air Forces asked some of the leading electrical manufacturers of the country to undertake the development of a wholly new alternating-current electric system for aircraft. While some of the details are not ready for publication, enough can be revealed to make this an opportune time to review the progress which has been made, and to evaluate the significance of this highly important development.

Fundamentally, the object of an alternating-current system for aircraft is weight economy through the use of higher voltages than now appear practical for direct-current systems. Preliminary studies of alternating-current electric systems started long before the war. Navy experience with high-power radio equipment had demonstrated as early as 1930 that alternating-current power equipment was far more reliable than direct-current equipment for supplying high plate voltages of the order of 1000 to 2000 volts. Use of dual generators providing variable-frequency alternating current for radio and direct current for the power system is common practice on Navy airplanes, and is

a compromise between direct-current and alternating-current systems which is adaptable to many airplanes of moderate size.

For large airplanes a higher system voltage than the present 27-volt value is essential. Searching for ways to increase voltage, the Army authorized two experimental airplanes equipped with alternating-current systems in the middle 1930's. The XB-15, shown in the head illustration (Boeing's "Old Grandpappy", 1937 forerunner of the B-29), was equipped with a single-phase 800-cycle system, and the XB-19 (Douglas' "Flying Laboratory") was equipped with a 120-volt, 400-cycle, three-phase system. System capacities were relatively small, the power supply for each airplane being a pair of small auxiliary-engine-driven alternators, and the installations were far short of the elaborate systems now being developed.

Polyphase System Proved Superior

The XB-19 was test-flown June 27, 1941, some four years after test flights of the earlier XB-15. Operation of the electric equipment on the XB-19 demonstrated the superiority of polyphase power for motors and for parallel operation of alternators, as contrasted with previous experience with the single-phase system. Before December 7, 1941, alternating-current-powered airplanes had been interesting experiments, but with the advent of war they assumed new significance. Perfected alternating-current systems, according to the more enthusiastic engineers at Wright Field, would mean that the electric system would no longer be a limiting consideration in determining the maximum size of an airplane.

During the dark days of 1942, Japan rapidly acquired all of the air-bases between Pearl Harbor and Australia. As the war developed, it began to appear high time to start developing an electric system which would overcome the limitations inherent in existing low-voltage systems; that is, by going to something of the order of 230-volt industrial practice.

Controversy Is Still Raging

Long-range airplanes must necessarily be high-altitude airplanes and it was decided that high-voltage direct current at high altitudes was a gamble that could not be risked, partly because of the unknown hazards of low arcing voltages and partly because of the known hazards (at that time) of excessive brush wear at altitudes above 30,000 feet. With this decision made, and a controversy initiated that is still raging, the Electrical Branch of the Equipment Laboratory at Wright Field set up an alternating-current engineering unit and attacked the problem of an alternating-current electric system in earnest. Tentative specifications were gradually whipped into shape. Basic elements of the system have been fixed since a general meeting at Wright Field on January 18, 1943. System characteristics are as follows:

Volts, line-to-line	208 at generator
	200 at load centers
Volts, line-to-neutral	120 at generator
Phases	3 phases
Frequency	400 ±20 cycles per second
Neutral	grounded

Basic generating units are main-engine-driven alternators rated 40 kva at 75 per cent power factor. They are designed for parallel operation when suitable speed-controlled drives are provided. Four such units provide 120 kw (160 kva) and require

distribution lines rated approximately 440 a
peres per phase. By contrast, the familiar "2
volt" system, operating at 27 volts, would requi
copper to carry 4400 amperes. For long co
ductor spans, where regulation rather th
thermal capacity determines the size of conduct
required, the single 27-volt conductor would r
quire 17 times the copper that is required for a
three of the 208-volt conductors. The reductio
in weight is due entirely to the change in volta
and has nothing to do with frequency phases.

Motors are of course polyphase, *Fig.* 1 sho
ing a typical rotary actuator. They are operate
grounded-neutral so that they will develop parti
ratings with one line shot away, and will co
tinue to run at light load with two lines sh
away, *Fig.* 2, although they will not restart und

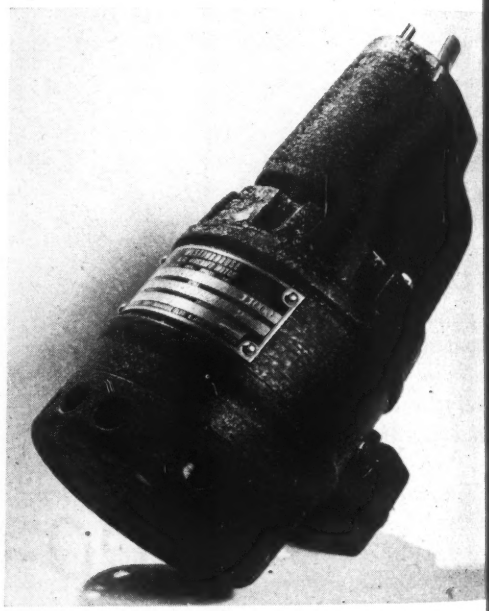


*Fig. 1—Above—Rotary actuator composed
400-cycle gearmotor rated 200 inch-pounds
27.5 revolutions per minute.*

*Fig. 2—Below—Speed-torque curves of a 3-phas
4-wire induction motor showing effect of havi
one or two circuit wires shot away*

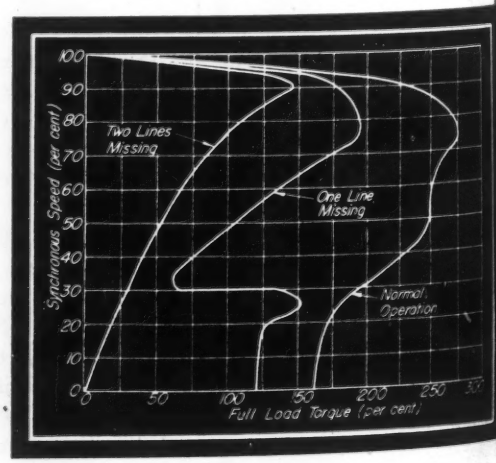


Table I—General Comparison of Various Electric Systems for Large Aircraft Generators Rated 24 to 32 Kw

Equipment	400-Cycle, 3-Phase, 208/120-Volt, 4-Wire A-C System			24-Volt Direct Current System	120-Volt Direct Current System	Dual Generator System 24-Volt, Direct Current 120-Volt, Single-Phase Var.-Freq. A-C
Prime Mover	(A) Main engine	(B) Auxiliary engine	(C) Auxiliary gas turbine	Main engine	Main engine	Main engine
(A) Weight	Light	Very heavy	Very light	Light	Light	Light
(B) Fuel efficiency	Good	Very poor	Very poor	Good	Good	Good
(C) Maintenance	Normal	Considerable	Minimum	Normal	Normal	Normal
(D) Availability	Available	Not fully developed	Not available at present	Available	Available	Available
Coupling	Variable-ratio drive	Step-up gears	Step-down gears	Double-universal jack shaft	Double-universa jack shaft	Double-universal jack shaft
(A) Weight	Probably 3-5 lb/kva	Low	Low	Low	Low	Low
(B) Efficiency	Probably 60-80%	95-97%	95-97%	Negligible losses	Negligible losses	Negligible losses
(C) Maintenance	Probably high	Slight	Slight	Negligible	Negligible	Negligible
(D) Availability	Not available	No problem	No problem	Available	Available	Available
Generator	High-speed (6000 rpm) alternator			Wide speed range (2700 to 9000rpm)largecommutator complicated brush rigging	Wide speed range (2700 to 9000 rpm) moderate commutator	Wide speed range (2700 to 9000 rpm) two generators in one housing
(A) Weight	Approx. 2 lb per kva (including exciter)			Approx. 5 lb per kw	Approx. 4.5 lb per kw	Approx. 6 lb per kw
(B) Efficiency	Approx. 90%			Approx. 85%	Approx. 88%	Approx. 85%
(C) Maintenance	Slight			Moderate	Moderate	Moderate
(D) Availability	Available for 40 kva at 75% pf			Not fully developed	Not fully developed	Not fully developed
Excitation	Integral d-c exciter (weight included in alternator weight)			Self-excited or separate exciter	Self-excited or separate exciter	Self-excited
Control	Complicated control required—will require trained technicians for maintenance			Simplest control	Control fairly simple—special interruption devices required	Dual control required
Speed Governor	Accurate speed control required for frequency control and for load division. Must operate more rapidly than engine speed can change			None	None	None
Voltage Regulator	Operates on exciter field. Must divide reactive power as well as control voltage. Weight 10-15 lbs			Requires a heavy-duty regulator unless an exciter is used. Weight 10-15 lb	Requires a heavy-duty regulator unless an exciter is used. Weight 10-15 lb	Requires two regulators: d-c regulator 5-12 lb a-c regulator 10 to 35 lb
Meters	Volts: Amperes: Kilowatts: Kilovars: Frequency			Volts: amperes	Volts: amperes	d-c volts, d-c amperes a-c volts, a-c amperes
Bus Connection	Synchronizing control: Frequency and voltage relays: Three-pole latched-in circuit breakers with d-c operating coils			Reverse current or differential-voltage relay	Reverse current or differential-voltage relay	Reverse current relay for d-c generator
Protective Devices	Differential phase relay: Exciter-ceiling relay: Three-phase breakers for isolation and transfer: Thermal protection: Low-frequency relay: Fuses or other circuit-protective devices			Generator fault breaker: Thermal protection: Fuses or other circuit-protective devices	Generator fault breaker: Thermal protection: Fuses or other circuit-protective devices	Generator fault breaker: Thermal protection: Fuses or other circuit-protective devices
Distribution	Parallel operation requires complicated control. Three relatively light-weight circuits are required. Fuselage used for neutral. Considerable reactive power must be circulated			Parallel operation is simple. One extremely heavy circuit required. Voltage regulation a serious problem. Fuselage used for return	Parallel operation is simple. One or two medium-weight circuits required. Some objection to use of grounded return	Parallel operation of d-c units is simple. Parallel operation of a-c units is impossible. One heavy circuit and two light circuits required
Conversion Devices	Light-weight transformers for a-c to a-c voltage transformation. Approx. 2 lb per kva. m-g sets or static rectifiers for a-c to d-c conversion. Weight about 10 lb per kw, efficiency about 60% at full load.			Large inverters or additional engine driven alternators required for radio and radar. 10 to 20 lb per kva. Dynamotors for radio 25 to 150 lb per kw	Large inverters or additional engine driven alternators required for radio and radar. 10 to 20 lb per kva. Dynamotors for radio 25 to 150 lb per kw	Light-weight transformers for a-c to a-c voltage transformation. Dynamotors for low power radio. Small inverters for constant-frequency instrument circuits
Motors	Characterized by high speeds, low weight, simplicity, minimum maintenance, moderate torque. Require at least two thermal protectors. Require dc for brakes and clutches.			Moderate speeds, moderate weight, high torques, moderate maintenance. Single thermal protector	Moderate speeds, weight for large motor slightly less than for 27-volt motors, high torques, moderate maintenance. Relays probably required for thermal protectors	Moderate speeds, moderate weight, high torques, moderate maintenance. Single thermal protector
Radio	Operate direct			Use dynamotors or inverters	Use dynamotors or inverters	Operate from a-c or from d-c with dynamotors
Turrets	Ideal for motor on variable-voltage m-g sets. Facilitates use of electronic computers. May be suitable for thyratron drive			Slightly heavier than a-c for variable-voltage m-g sets	Slightly heavier than a-c for variable-voltage m-g sets	Slightly heavier than a-c for variable-voltage m-g sets
Instruments	Ideal for gyroscopes and synchro-tie equipment. Facilitates design of many other instruments			Usually require inverters	Usually require inverters	Usually require inverters
Control Surface Actuators	Ability to synchronize, a very important consideration			Flap controls, etc., require means for synchronizing	Flap controls, etc., require means for synchronizing	Flap controls, etc., require means for synchronizing
Lighting	Easy to use most favorable voltage			Satisfactory	Believed satisfactory	Satisfactory
Heating Devices	Satisfactory			Satisfactory	Satisfactory	Satisfactory
Cabin Supercharging	Does not appear practical because of need for speed control			Possible but weight excessive. Power requirements would be very heavy	Possible. Weight probably higher than other methods	Possible but weight excessive. Power requirements would be very heavy

the latter condition.

Availability of alternating-current systems does not mean that all large airplanes will or should use them. In selecting the system best suited to a particular application many factors must be considered, the size and electric power requirements of the airplane being two of the most important. Installation weight, fuel consumption and, for a military airplane, the probability of an airplane reaching its destination, must all be considered. Fuel consumption for operating equipment may be of far more consequence than the equipment weight for operation over extreme ranges.

Reliability is fully as important as weight. Pay-load capacity achieved at the cost of reliability is not a good investment; capital would be expended faster than the fictitious high rate of return could ever justify. Where possible, of course, the designer will provide the spares necessary to assure the return of the airplane even though some vital device fails because of enemy action or because of its inherent weakness. The weight of this reserve equipment must be subtracted directly from pay-load capacity. Remaining deficiencies in reliability must be made up by supplying additional airplanes and crews.

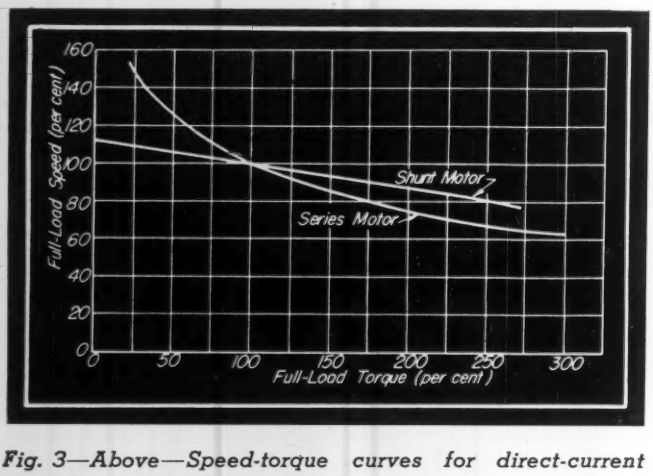
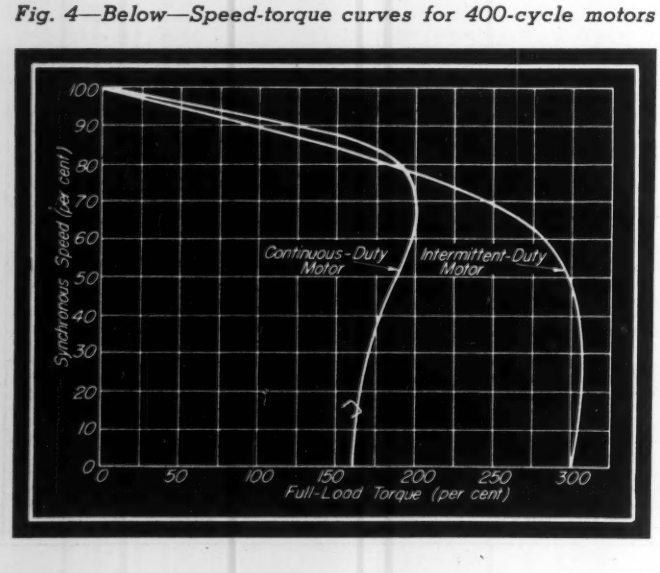


Fig. 3—Above—Speed-torque curves for direct-current aircraft motors. One is for shunt, other for series motors

Fig. 4—Below—Speed-torque curves for 400-cycle motors



Conversely, an improvement in reliability cuts do losses. "All-electric" airplanes, according to the AAF, much less vulnerable to combat damage than are hydra ically equipped airplanes. This is one of the reasons the military importance of electric systems, since t transition from hydraulic to electric operation of actuat and accessories is a major factor in the expansion of el tric power requirements on large airplanes. The B-29, though equipped with a 24-volt, direct-current system, considered the first all-electric airplane, and has the lar est electric system capacity of any airplane regardi which information has been released.

Theories Are Not Yet Proved

Neither comparative reliabilities of 24-volt and 120-v direct-current equipment, nor the comparative reliabiliti of 120-volt direct-current and 208-volt alternating-curre equipment have yet been established by operating expe ence, although some experimental and much theoretic evidence is available on these subjects. In general, t absence of brushes and commutators on motors will i crease reliability, and it is generally admitted that circu interruption problems will be less with alternating curre than with direct current at comparable voltages. On t other hand, it now appears that a variable-ratio drive, least for some time to come, may be as much a hazard reliability as were brushes at high altitudes at the time t original decision was made to develop an alternating-cu rent system.

Closely related to reliability is the factor determini the time an airplane will be grounded for maintenanc This factor may be evaluated as the pay-load utilizatio factor. As the intricacy of the equipment increases, t time for ground maintenance is likely to increase in pr portion. Some of the equipment proposed for alternating current power systems is admittedly much more intricat than equipment now being used for 24-volt systems. To much should not be made of this point, however, as 24 volt equipment is also growing more intricate as new re quirements develop.

Although actual service will tell the final story, som predictions regarding maintenance have been made fo actual or anticipated designs on the basis of such gener considerations as the number of parts involved, the nece sity for abnormally accurate tolerances, likelihood of ex cessive temperatures, vulnerability to combat damage, an ability to withstand abuse and improper operation. Eva uated on this basis, constant-speed drives and synchroni ing controls appear likely to require somewhat greate maintenance and a somewhat higher standard of technic ability to operate them than do any components of t existing 24-volt systems. Motors for use with alternating current systems, on the other hand, will require conside ably less maintenance than direct-current motors now r quire.

So far, electric systems have been discussed rather ge erally as to the effect that the system may have on the ai plane. For proper overall perspective it is now necessar to consider the fact that all large airplanes carry som equipment which must be operated from direct curren

some which must be operated from alternating current. Solenoids and some forms of control apparatus function best when designed for direct current, whereas many instruments operate from alternating current. Regardless of the type of power provided for the main system, conversion equipment will be required to suit many of the devices to be operated. Where possible, of course, devices will be designed to operate from the basic power supply without the use of conversion equipment.

Table Compares System Components

Although an alternating-current motor will weigh about 50 per cent as much as an equivalent direct-current motor, it will not pay to install an inverter to permit the use of alternating-current motors on direct-current equipped airplanes unless compelling reasons other than weight exist. On the other hand, radar tubes require power at a direct-current potential of many thousand volts. The lightest and most reliable method of obtaining this power is by transforming high-frequency, alternating-current power to a suitable voltage and then rectifying it to provide the required direct-current potential. The weight required on direct-current airplanes for inverters or for special alternators provided solely for radar operation is considerable, but it is negligible compared with that required for direct-current generation at the necessary voltage.

Comparison of complete electric systems is facilitated by breaking the system down to the following elements:

1. Prime mover
2. Coupling device (with or without speed ratio)
3. Generator
4. Excitation source
5. Control and protective devices
6. Distribution circuits
7. Conversion devices
8. Utilization devices.

These elements may be compared item for item for various systems and for all sorts of airplanes. An attempt has been made to indicate the salient characteristics of the components of various systems in TABLE I. In such a table, remarks are necessarily limited to generalities, but a great deal of detailed information is available in technical publications for those who require it.

With so much progress in developing equipment for alternating-current systems already recorded, it is necessary to report that one tremendous obstacle has not yet been overcome. In MACHINE DESIGN for June, 1941, Dr. E. E. Minor discussed the subject of "high-frequency power for aircraft" and pointed out the need for a constant-speed device to couple the alternator to the main engine. Not much more can be said in 1945, at least at this writing. The variable-stroke, friction-drive device to which Dr. Minor alluded was abandoned some time ago, but there are at least three hydraulic devices which bear some promise of success, although at much higher weights than were originally contemplated. Recent developments in gas turbines also suggest that an auxiliary gas turbine may be a suitable prime mover in spite of high fuel consumption, and that development of such a turbine may make it possible to abandon the long quest for a constant-speed drive.

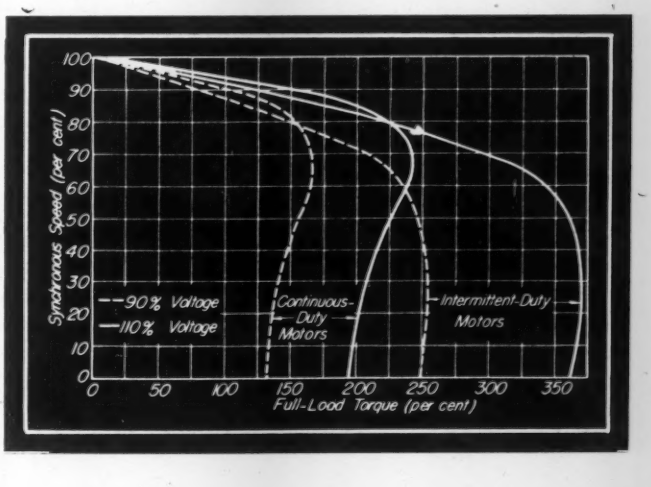
TABLE I indicates that alternating-current systems will be justified for high values of connected load, and for airplanes of large physical size with power-transmission distances exceeding 100 or possibly 200 feet. The dividing line between preference for direct-current systems and preference for alternating-current systems is not too clearly established, and will shift somewhat according to the success in developing an alternator drive and, as pointed out previously, according to the particular make-up of the various utilization devices which will be required on the airplane.

One question of paramount importance cannot be answered at this time, namely: When will the new alternating-current system be in actual operation? Some of the urgency has disappeared since 1942. Then again, much of the early impetus came from the high-altitude brush problem, a major obstacle to direct-current apparatus in 1942, now almost a forgotten problem thanks to the work of Dr. Howard Elsey and his associates. Even so, with so much of the equipment already completed, it does not appear that flight tests will be long delayed, although it is admitted that the first installations cannot be as complete as proponents of the alternating-current system would like to see them.

If this prediction is correct, it behooves the designers of aircraft accessories to be ready with alternating-current versions of aircraft electric equipment. In general, conversion of existing direct-current motor-driven devices will be relatively simple and, in most cases, the alternating-current device will be simpler, lighter and easier to maintain than its direct-current counterpart. Often, an alternating-current motor will merely be substituted for the direct-current motor, with negligible modifications of the accessory. Many designers, however, will want to take advantage of the possibilities inherent in alternating-current motor application by redesigning the accessory for a higher motor speed.

The most popular speed for direct-current motors is prob-

Fig. 5—Effect of voltage variation on speed-torque characteristics of continuous-duty and intermittent-duty 400-cycle motors. Voltages are for maximum limits permissible on aircraft



ably 7500 revolutions per minute; 11,500 revolutions per minute is expected to be the most popular speed for 400-cycle motors, although conservativtve designers may hesitate to exceed 7500 revolutions per minute except for intermittent ratings or for continuous ratings below one horsepower. Very large motors are likely to warrant special lubrication systems which will permit operation at 11,500 or 23,000 revolutions per minute, speeds which are already exceeded on gas-turbine compressors and on superchargers.

Use of built-in alternating-current motor parts is considerably simpler than is the case with direct-current motor parts, since no brush rigging is used. As a compromise between motor parts and complete motors, the AAF are sponsoring the development of "three-quarter" motors, the bearing housing for the drive end being a part of the accessory instead of the motor. This method of construction saves some weight compared to complete motors, and facilitates motor replacement compared with integral motor parts, but is being resisted by many manufacturers because it complicates problems of ventilation, factory testing, and responsibility for satisfactory operation.

Care must be taken in applying any motor to recognize its speed-torque characteristics. For this reason typical speed-torque curves for polyphase motors are shown and contrasted with the speed-torque characteristics of direct-current motors in *Figs.* 3, 4 and 5. These curves are for illustrative purposes only, and the manufacturer's curves for a particular motor must be used for design purposes. Fear has been expressed that alternating-current motors with high-ratio gear reductions may not have sufficient starting torque to overcome the drag of congealed lubri-

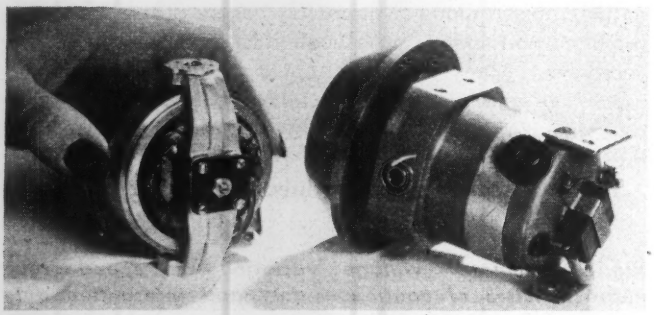


Fig. 6—400-Cycle hysteresis-synchronous gyroscope left, compared with direct-current motor-driven gyroscope. The former operates at 12,000, the latter between 7500 and 10,000 revolutions per minute

cants at minimum ambient temperatures, but tests have shown that such fear is groundless if motors are properly designed.

Designers who are familiar with portable tools know that the use of high-frequency, polyphase motors for high speeds and for high power at low weight is well established for grinders, woodworking tools and for many other hand-held tools. Such motors have no brush nor commutator maintenance problem, a serious obstacle to the production-line use of some portable tools with series-wound, universal motors.

High-power output with low weight is not the only desirable characteristic of high-speed alternating-curre motors. Gyroscopic instruments are already largely alte nating-current-motor driven, although there are still ma instruments using direct-current gyros. Most of the would be improved by the substitution of alternating-cu rent motors, since these motors have no brush friction, a free from carbon dust to spoil precision bearings, are easi to balance, and are much more uniform in speed. Th hysteresis-synchronous motor, *Fig.* 6, is particularly sui able for small gyros as it combines accurate speed wi simple construction. An interesting fact in favor of th type of motor is that it has a relatively constant impedan from locked-rotor to synchronous speed.

Use Power Best Suited to Device

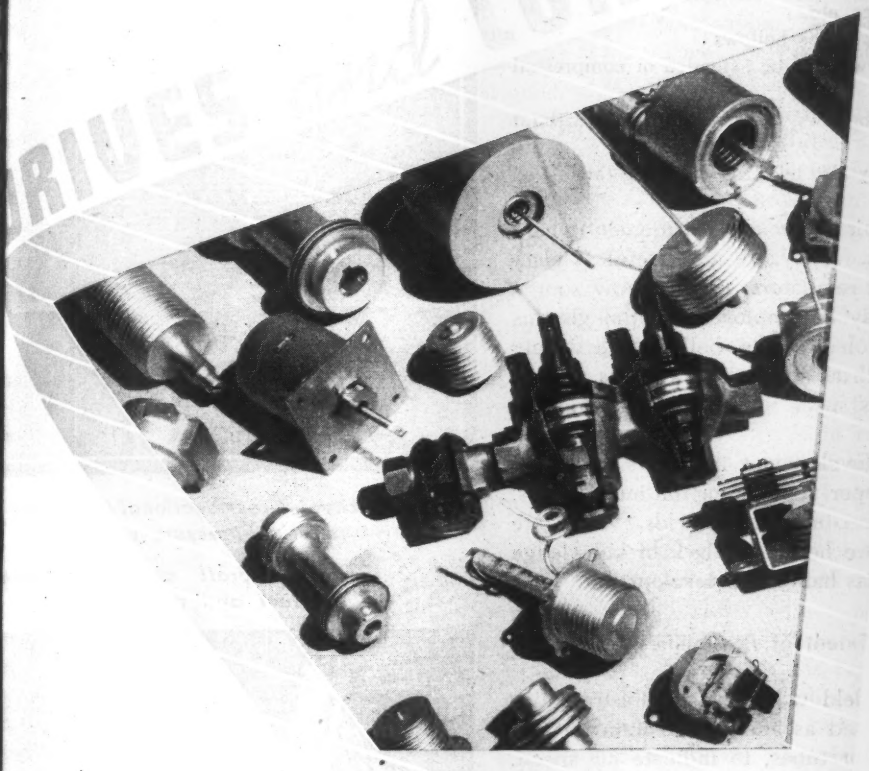
When alternating current is available, fluorescent lamp ultraviolet instrument illumination, and filament typ lamps may be employed where each is most suitable, an each type may be operated at its most favorable voltag

Radio equipment is readily designed for alternatin current operation and, in fact, the majority of high-pow ered transmitters are already constructed for single-phas high-frequency power. Radio may be operated sing phase as at present by using line-to-neutral power or ma be redesigned for polyphase operation with some reduc tion in weight of the power-supply filter. Although a co stant frequency is not necessary for the plate-voltage sup ply, it is frequently necessary for some devices. This one advantage of the new system as compared with som installations where variable-frequency power is now pro vided by means of variable-speed alternators; another ad vantage is that far greater reserve capacity is available i case one or two alternators fail.

Operation of low-power radio transmitters and receiver from alternating current eliminates the need for rotatin dynamotors for plate power but it also eliminates the pos sibility of emergency operation from battery power. De pendence for emergency power may be placed upon a auxiliary alternator set, or some emergency radio equip ment may continue to be operated from the 28-volt bus.

Alternating-current motors appear to suffer by compari son with direct-current motors when variable speed is re quired. The principal use, however, for variable-speed motors is for turret operation, and these motors must op erate with precise control from zero speed to maximum speed in either direction. This is now accomplished b using direct-current, variable-voltage motors with fixed field excitation. A variable-voltage generator and gen erator-driving motor must be provided for each variable voltage motor. Present systems may be used almost in tact by driving the variable-voltage generators from a ternating-current motors instead of from direct-curren motors. Considerable reduction in weight is possible b driving two generators from one motor, since both gen erators never demand full power at the same time. Maxi mum motor rating occurs when azimuth and elevation speeds each equal 70 per cent of the maximum turre speed. Field excitation for the azimuth and elevation motors may be obtained from small static rectifiers, o

(*Concluded on Page* 190)



Where Deflection by Pressure Is Required!

By William F. Laser
Manager, Bellows Division
Cook Electric Co.

METAL bellows, while simple in appearance, require precise specification because of their varied, unusual and difficult functions. For instance, in the illustration above are shown a few typical applications including switches, heat motors, couplings, valves, pistons and regulators. Today, metal bellows are performing functions which a few years ago had been termed by engineers as not only impractical, but impossible. What engineers can expect from bellows, what jobs they perform, and what consideration must be given them in design is the subject of this article.

There are eight physical and operating characteristics which comprise the general terminology when discussing a bellows. These are as follows:

1. Outside diameter: Measurement across the end of the bellows from one outside edge to the other
2. Inside diameter: Measurement from one inside edge to the other or, in other words, the

distance across the hole in the center of the bellows

3. Free length: Height of the bellows measuring from one end to the other while in a relaxed position, that is, while not subjected to any strain or pressure
4. Kind of metal, its thickness, temper, etc.
5. Number of convolutions or flanges in the bellows
6. Travel, or the amount that a bellows may be extended or compressed without injury
7. Maximum pressure to which a bellows may be subjected without injury
8. Flexibility of a bellows, which may also be termed "spring rate".

Bellows can be designed to suit almost any requirement where deflection by pressure is required. They are also useful as electronic wave guides or as pressure regulators, *Fig.* 1. Any source of pressure—air, gas or fluid—may be employed. Although this article deals primarily with control elements, bellows are widely employed as sealing members both as stators or rotors, as permanent flexible couplings, as well as shock mounts or vibration absorbers.

During the present era of development in which automatic controls are essential for the proper functioning of mechanisms, bellows have been adapted in ever-broadening fields. They are being applied in many cases where heretofore lack of knowledge of bellows and their possibilities has hampered development.

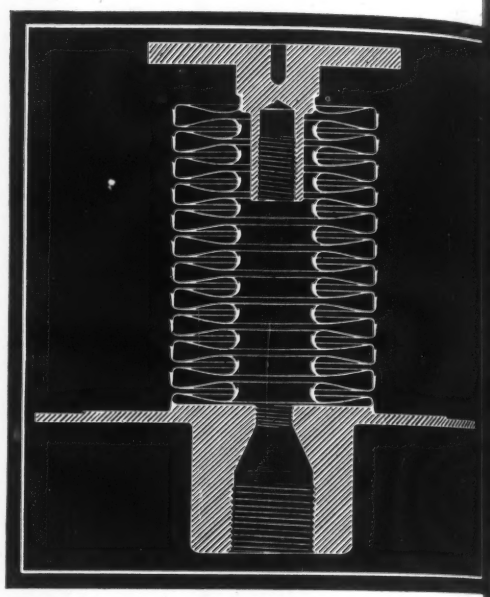


Fig. 1—Above—Cross-sectional diagram of typical oil-pressure regulator

Fig. 2—Below—Aircraft altitude compensa for fuel and air mixture

Applications Indicate Extent of Possibilities

Applications in the aircraft field will serve to illustrate this point. Primarily, bellows were used as aneroids to indicate altitude and, in connection with pitot tubes, to indicate air speed. Now, bellows are employed to indicate manifold, fuel and oil pressures. In conjunction with a carburetor as engine temperature and altitude measuring units, they control the mixture of fuel and air to each engine, *Fig.* 2. They also regulate the proportion of water injection with relation to manifold pressures and altitude. Pressurized cabins utilize them to regulate the required amounts of oxygen at the proper altitudes. Bellows pressure switches, *Fig.* 3, may be used to give signals or to control associated equipment at any desired pressure.

Jet-propelled planes have many possibilities for application of bellows as control units. New problems associated with the advanced design of these planes have resulted in the development of bellows that will withstand the higher temperatures and pressures met in jet-propulsion service.

In considering whether a bellows will solve a specific problem it may be borne in mind that bellows are not limited with respect to size, and that problems of corrosion are readily overcome. Metals such as beryllium copper, phosphor bronze, brass, stainless steel, special alloy steels, monel and inconel have been employed. These, when properly heat-treated in the bellows form, provide a previously determined spring characteristic and are capable of withstanding excessively high pressures or temperatures.

Methods of metal joining used in the construction of bellows include processes such as controlled-atmosphere copper brazing, atomic-hydrogen welding, heliarc welding, silver brazing and induction brazing.

Many engineers can design a bellows for their own application. It is wise, however, to call in a bellows engineer for consultation when contemplating the use of bellows in any new development. Here are the points that must be considered in specifying a bellows:

1. Maximum external or internal operating pressures
2. Differential pressures
3. Maximum or minimum inside and outside diameters
4. Displacement or volume
5. Thrust force
6. Maximum or minimum spring gradient
7. Total deflection in extension or contraction
8. Free length
9. Lateral deflection
10. Maximum and minimum temperatures
11. Operating medium, internal or external
12. Bellows material
13. Maximum vibration

4. Type of attachments, top and bottom
5. Kind of finish.

Life of a bellows and its performance depends upon its ration within maximum operating limits. These limits defined by the travel ratio wherein the deflection is ar. A bellows cannot be selected until the maximum rity is found and a suitable safety factor is assumed. must be understood that under "no load", where the se is linear, the life expectancy of a bellows is in-nite. However, upon increasing pressures in a non-ar course, the life is reduced.

Bellows with high spring gradient are selected for high ssures. Therefore, a lesser deflection per increment of ssure may be expected. For lower pressures, a low ng gradient may be employed to give a greater move-nt per increment of pressure. High pressure would rupture a bellows with a low spring gradient, where-a low pressure would deflect a high-pressure bellows a y small amount.

Too often a problem is approached with the bellows ing the final consideration. In so doing, limitations are ced on bellows selection, size, area or free length. De-gners should not expect a small capsule of a bellows to flect a great distance, a light-gage metal of being capable withstanding extremely high pressures, or a high-pres-e unit to deflect a great distance. This sometimes neces-tes a complete redesign of a device to accommodate proper unit. The engineer should remember that a lows, while a flexing unit, must operate within its nal space requirements. An indication of the possi-ties of bellows' extension is illustrated in *Fig.* 4, where length of 30 inches is twice the compressed length. In most devices where a bellows is used, it is the heart the device itself and should be given primary consid-eration. It should be the nucleus around which the entire unit is built. While "impossible" applications, in most instances, can be solved, they create an added amount of engineering and manufacturing cost. Often this results in designers saying "a bellows won't do", when in reality they have not taken all factors into consideration and given it a fair chance. Again, they will find that they must be satisfied with an approximation of the perfection that could have been had, had they considered bellows engineering problems from the beginning.

Consideration that must be given the selection of a bellows when building gages may serve as an example. All operating factors of the gage must be determined, including pressures, size of final instrument, linearity for accurate indication, minimum deflection required, methods of adjustment, methods of mounting, maximum hysteresis. Dependability in operation and life expectancy are of utmost importance. In many types of electrical gages the measurements are made in millionths of an inch. Hysteresis then becomes one of the most important factors. It is generally known that regardless of how accurately an indicator is built, there is always some loss between increasing and decreasing pressures. Electrical indication reveals this characteristic in many instances. A highly tempered bellows with a small deflection is most properly adapted for minimum hysteresis. As an example, an electrical instrument has been built recently wherein twenty millionths of an inch was considered high hysteresis on a travel of one-tenth-inch. In the selection of a bellows of high-tempered phosphor bronze this hysteresis was reduced to a maximum of five millionths. An engineer in the development of critical instruments should specify that the bellows required be calibrated through to spring gradient or deflection per given increment of pressure, within certain limits of hysteresis.

Generally, the best performance is obtained in any bellows as an indicating device when pressures are employed externally. This is attributed to the fact that travel is more linear in compression than it is in extension, and that more than double the pressure can be applied when pressures are employed in this manner.

There are different types of bellows construction; the rolled type, the hydraulically formed type, the helically formed type and the joined diaphragm type. Some are manufactured specifically with economic consideration toward large production. Special types for operating precision that have greater expectancies in life and low hysteresis are naturally more costly.

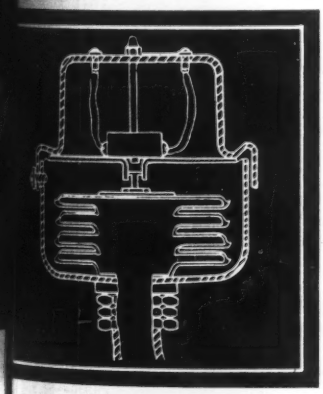


Fig. 3 — Left — Bellows incorporated in switch permits operation at the desired pressure

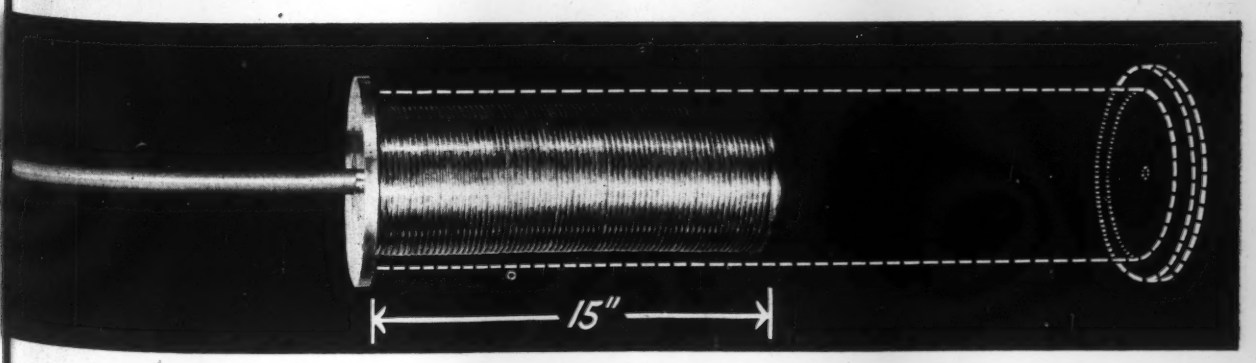


Fig. 4—Below— Special bellows, extended, is twice its compressed length of 15 inches

Single-Line Air Cylinder Control

By Spencer B. Maurer
Research Engineer
Chicago Pneumatic Tool Co.

SIMPLICITY of design is emphasized in a recently developed remote control for an air cylinder. This control uses only one air line leading to a conventional shutoff valve, and regulates the flow of refractory powder and air in a "gun" used to apply a refractory mix to the hot vertical walls of steel-making furnaces.

The gun unit on which the control is employed consists of a hopper, a 20-foot length of two-inch diameter reinforced rubber hose, air and water lines, and a mixing nozzle and discharge pipe. It is mounted on wheels for convenient mobility. The mixing nozzle and discharge-pipe unit, fed by the long flexible lines, is held in the operator's hands and manipulated so as to direct the discharging mix properly on the furnace walls. Because it is first necessary to chill slightly and clean the walls with an air and water stream, the function of the control must be to shut off the powder while maintaining the air and water feed.

In *Fig.* 1 is shown a cross section through the hopper and ejector unit. The air jet is mounted in a movable plunger which is moved to the open position (shown) by pressure in the air cylinder, and is closed by the force of the springs when the air pressure is relieved. When closed, the rubber seal at the end of the air-jet plunger is wedged into the entering taper of the ejector tube, forming a tight seal which prevents leakage of the refractory powder into the discharge stream. When open, the powder is fed by gravity through the hopper down to the active area of the jet where it is picked up by the air stream and carried out through the discharge hose and mixing nozzle where the correct amount of water is injected into the stream.

In the conventional method of controlling such an air cylinder—illustrated in *Fig.* 2—two hoses are required from the hopper unit to a two-way control valve at the mixing nozzle. One of the two hoses is connected to the air supply and the other to the air cylinder. With the valve set as shown at the left, the cylinder would open. With the valve plug rotated 90 degrees, the line air would be shut off and the air cylinder opened to the atmosphere, allowing the air-jet plunger to close off the hopper.

With the "single-line" control of *Fig.* 1, the air inlet line is eliminated and the two-way valve replaced by a simple petcock. Air is supplied to the cylinder from the plunger through a small hole. Exhaust from the air cylinder is through the control line and petcock. By closing this petcock, air pressure is built up in the air cylinder and the air-jet plunger retracts, opening the hopper feed. To close off the hopper feed the control petcock is opened, allowing air to escape from the cylinder more rapidly than it enters due to the relative sizes of the inlet hole and the exhaust air line.

Fig. 2—How conventional control operates

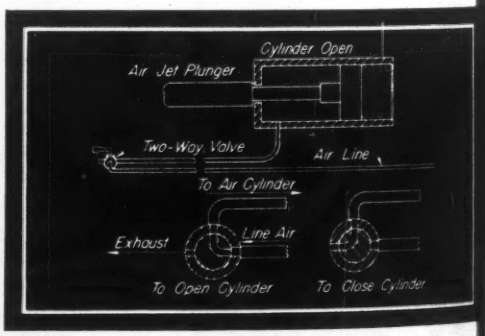
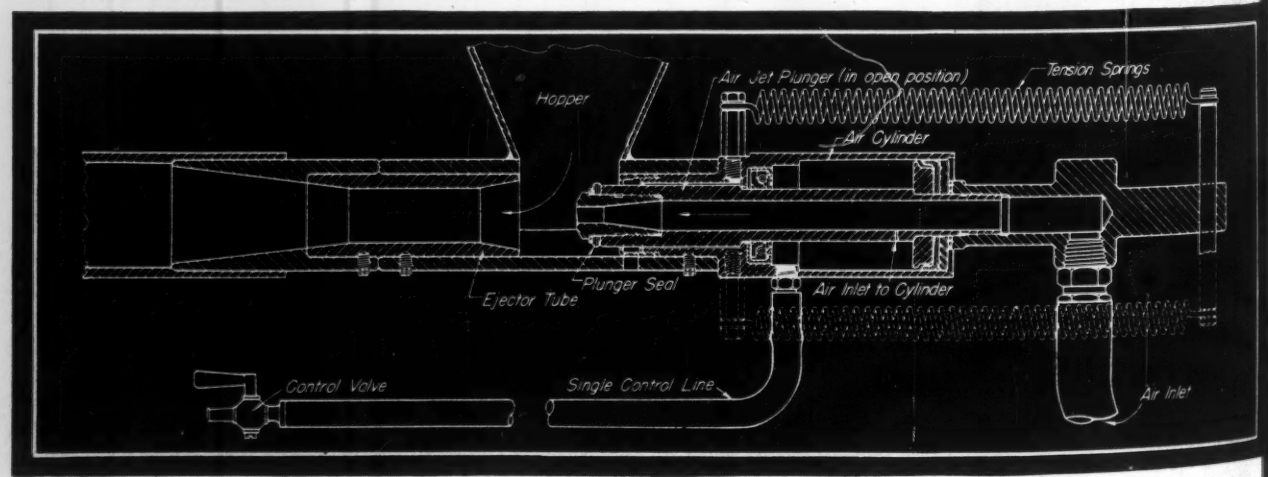
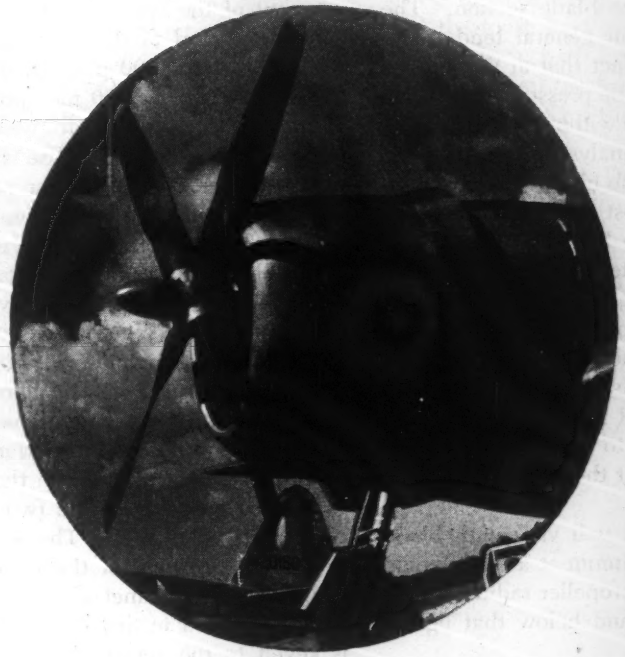


Fig. 1—Below—Drawing shows how air is fed directly through plunger to jet and cylinder. Cylinder closes when control valve is opened





How Planetary Reducer Aids Compact Design

By M. E. Cushman
Project Engineer, Propeller Division
Curtiss-Wright Corp.

IN ADDITION to its basic function of efficiently converting engine torque to useful thrust, the modern controllable-pitch propeller is expected to perform several other functions essential to the safety and efficiency of modern aircraft. Of these, first in importance is the constant-speed control required of the propeller in order to provide optimum powerplant efficiency. Associated with this requirement is the desirability of governing and synchronizing speeds of the engines of a multiengined airplane within approximately one revolution per minute. Also, the ability to feather is a necessity for propellers used on multiengined airplanes, to avoid windmilling and to reduce drag when one engine becomes inoperative for any reason. Ability to reverse the blade angle to negative values to produce a negative or braking thrust is a function proved desirable, particularly on multiengined airplanes, for use as a landing brake.

For the propeller to perform these functions, the pitch-changing mechanism is required to develop sufficient torque and power to move the blades promptly in response to signals from the controls. Because of the necessity for extreme compactness and

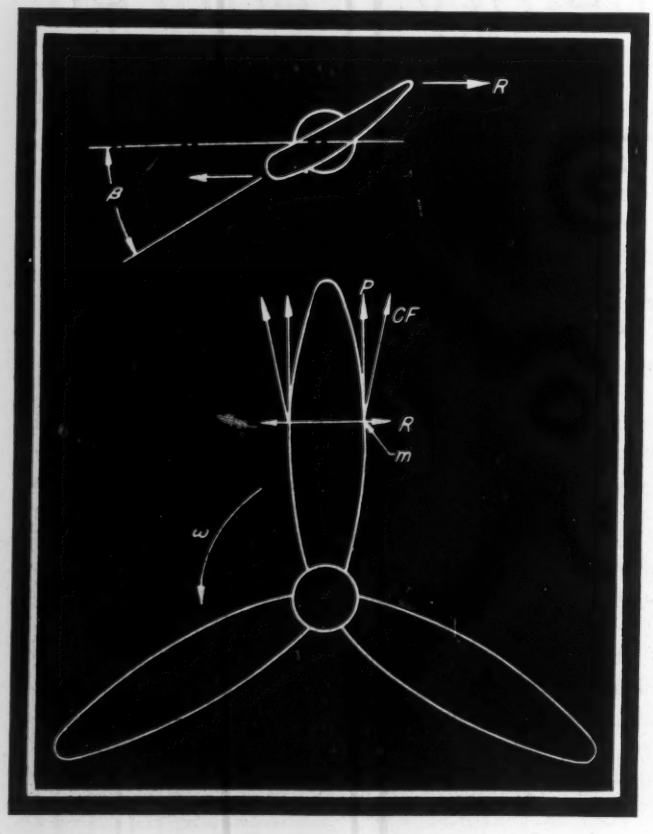
light weight, design of the unit presents unusual problems which it is believed will interest designers in general. The more significant of these will be discussed in this article, with particular emphasis on the design of the planetary speed reducer employed in the Curtiss propeller.

Drive for the moving of the blades must overcome inherent blade-twisting forces as well as the blade-retention friction. Blade-twisting moments are caused by air loads and by centrifugal action due to the blade section. The air loads are relatively small and in general tend to increase the blade angle, due to the fact that at normal operating angles of attack the center of pressure is forward of the center of rotation of the blade within the hub. This effect is usually disregarded in the analysis.

Centrifugal twisting moment due to blade section is definitely not negligible and can be most easily conceived by considering a single element of the blade and representing this element as two concentrated masses, *m*, at the leading and trailing edges of the section, *Fig.* 1. Direction of the centrifugal force, *CF*, on these masses is, of course, radial with respect to the shaft. This radial force can be resolved into two components *P* and *R*, where *P* is the direct tension component and *R* is in a direction tangential to the propeller axis. Due to the blade angle, β, component *R* causes a moment about the axis of the blade, tending to reduce this blade angle.

This centrifugal twisting moment will vary with blade angle in sinusoidal form from a maximum at a blade angle of approximately 35 degrees at .75 propeller radius to zero at blade angles 45 degrees above and below that figure.

Fig. 1—Below—Centrifugal forces produce torque tending to turn propeller blade into plane of rotation



At its maximum value, this twisting moment is about 1[illegible] 000 inch-pounds per blade for a typical 12-foot diame[illegible] propeller.

Friction of the blade-retaining bearings also must [illegible] considered when determining the total torque load on t[illegible] pitch-changing mechanism. With the angular-contact b[illegible] bearings used in the Curtiss propeller, this friction torq[illegible] in inch-pounds is approximately equal numerically to o[illegible] per cent of the bearing thrust load in pounds. Since t[illegible] centrifugal load of the blade applied to the bearing is [illegible] the order of 150,000 pounds, the resulting friction torq[illegible] is therefore about 1500 inch-pounds.

Sum of the centrifugal twisting-moment and bearin[illegible] friction torques for one blade is multiplied by the numb[illegible] of blades and the bevel-gear ratio to determine the to[illegible] pitch-change mechanism torque load.

The electric pitch-changing mechanism, consisting o[illegible] reversible direct-current motor coupled to the blad[illegible] through a speed reducer, provides the torque require[illegible] Constant engine speed under varying flight conditions [illegible] maintained by automatically adjusting the angles of t[illegible] blades, the pitch-change motor being energized in r[illegible] sponse to signals from an engine-driven governor mount[illegible] on the engine nose. Proportional correction of off-spe[illegible] conditions is incorporated in the governor design.

Speed reducer is of the two-stage compound epicyc[illegible] type, *Figs.* 3 and 4. The electric pitch-change mot[illegible] drives the sun gear of the input stage through a splin[illegible] coupling. Two planet gears mounted in a two-piece ste[illegible] spider are used in this stage. The larger, fixed, ring ge[illegible] is keyed to the speed reducer housing and the sun ge[illegible] for the low-speed or output stage is machined integ[illegible] with the movable ring gear for the high-speed stage. Fo[illegible] planets are used in the output stage of the unit shown [illegible] *Figs.* 3 and 4, for maximum load capacity, the plane[illegible] being mounted in a single-piece cast aluminum all[illegible] spider. Three planets are used in some units of oth[illegible] sizes. The movable ring output gear of the speed reduc[illegible] is splined directly to the bevel "power" gear which mesh[illegible] with and drives the blade gears as shown in *Fig.* 2.

This basic speed reducer design permits extremely hig[illegible] reduction ratios with minimum weight and maximum ef[illegible] ciency. Such units now in production have ratios var[illegible] ing from 7056:1 up to 9576:1, the one shown in *Figs.* [illegible] and 4 having a ratio of 7056:1. Units have been bu[illegible] for propeller use with reduction ratios up to 22,446:1 a[illegible] it appears entirely practical to provide at least 188,400:[illegible] in the same size housing simply by changing the num[illegible] bers of teeth on the planets and movable ring gears. R[illegible] duction ratio, *R* of one stage of such a speed reducer [illegible] given by the formula:

$$R = \frac{2T_{pf}T_{rm}}{T_s(T_{pf} - T_{pm})} \qquad (1)$$

where T_s = Number of teeth on the sun gear
T_{rf} = Number of teeth on the fixed ring gear
T_{rm} = Number of teeth on the movable ring ge[illegible]
T_{pf} = Number of teeth on large side of planet
T_{pm} = Number of teeth on small side of planet

For the speed reducer shown in *Figs.* 3 and 4 the r[illegible] duction ratio calculation is as follows:

$$R_{input\ stage}=\frac{2\times54\times109}{12(54-43)}=89.1$$

$$R_{output\ stage}=\frac{2\times18\times44}{10(18-16)}=79.2$$

$$R_{overall}=79.2\times89.1=7056$$

Efficiency of such compound epicyclic speed reducers is gh, probably higher for equivalent reduction ratios than ven by any other type of speed reducer of similar com- ctness. A relatively simple method for calculating the ficiency of this type speed reducer is derived in the fol- wing. Notation is, for any one stage:

TL = Tooth load at the output movable ring gear
R = Total reduction of the stage
R_s = Pitch radius of the sun gear
R_{pf} = Pitch radius of planet meshing with fixed ring
R_{pm} = Pitch radius of planet meshing with movable ring gear
R_f = Pitch radius of fixed ring gear
R_m = Pitch radius of movable ring gear
N_c = RPM of planet cage for one RPM output.

Reduction, R, and planet cage speed, N_c, are:

$$R=\frac{2R_{pf}R_m}{R_s(R_{pf}-R_{pm})} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (2)$$

$$N_c=\frac{R_{pf}R_m}{(R_{pf}-R_{pm})\ (R_s+R_{pf})} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (3)$$

Power loss at any gear contact is equal to some factor f times the tooth load times the velocity of engagement of the gears. The factor f is between .01 and .02, corresponding to 99 or 98 per cent efficiency for spur gears mounted on fixed centers. Simplifying assumptions are that losses at the sun gear are negligible due to the low tooth load and losses at both ring gears are equal. Then, using the velocity of engagement at the fixed ring gear and the tooth load at the movable ring gear (the greater of the two in both instances where the movable ring gear is smaller than the fixed ring gear) the following equation may be written for the efficiency, e, of one stage:

$$e=\frac{\text{Output}}{\text{Output}+\text{Losses}}$$

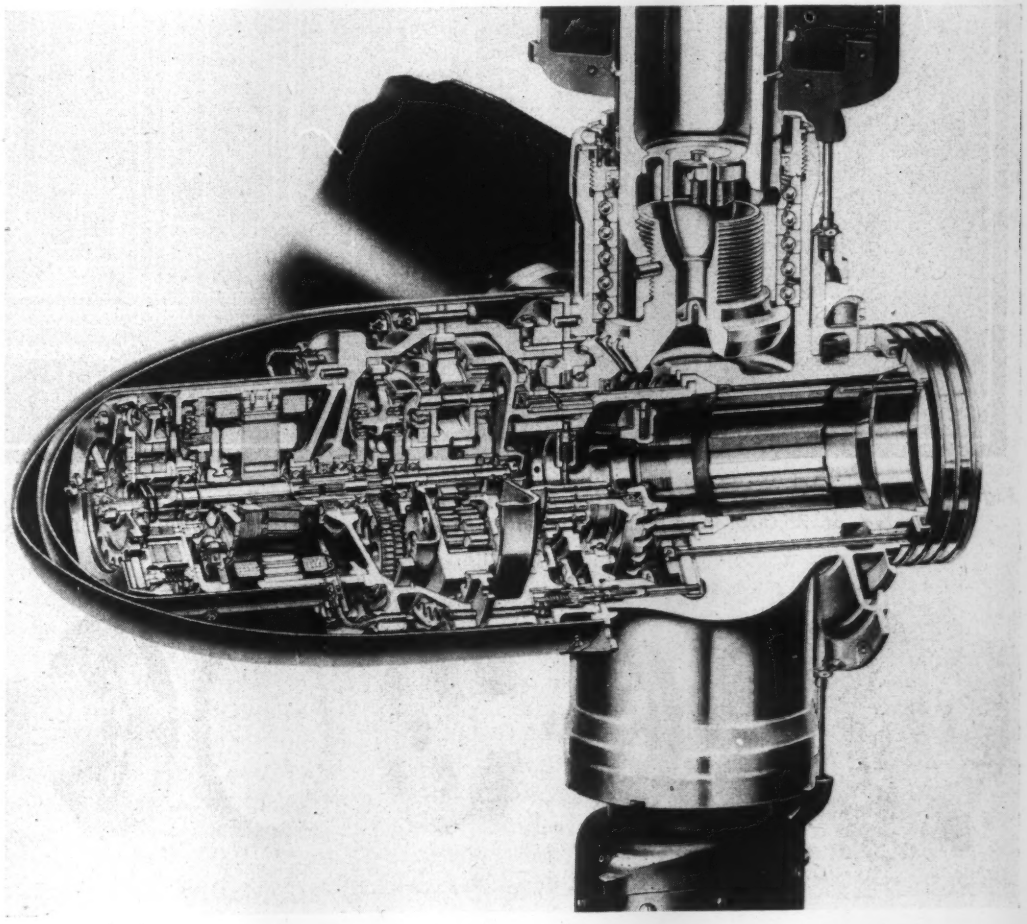
$$=\frac{TL\times1\times2\pi R_m}{(TL\times1\times2\pi R_m)+(2f\times TL\times N_c\times2\pi R_f)}$$

$$=\frac{1}{1+\dfrac{2f\times N_c\times R_f}{R_m}}$$

Substituting for N_c from Equation 3

$$e=\frac{1}{1+2f\left[\dfrac{R_{pf}R_m}{(R_{pf}-R_{pm})\ (R_s+R_{pf})}\times\dfrac{R_f}{R_m}\right]}$$

To simplify the expression in the bracket, it is first multi-



g. 2—Cutaway view electric propeller ows motor and two- ge compound plane- ary speed reducer

plied by R and divided by the value of R from Equation 2. Then the expression $2(R_s+R_{pf})$ is replaced by its equal $R_s + R_f$ (from the fact that R_f must equal $R_s + 2R_{pf}$) and the result is:

$$e=\frac{1}{1+f\left(\frac{2R}{R_m}\right)\left(\frac{R_sR_f}{R_s+R_f}\right)} \quad \ldots\ldots (4)$$

It will be noted that the numbers of teeth may be substituted for the pitch radii in this equation as long as the diametral pitch of both fixed and movable ring gears are the same. The assumptions made here are most nearly true when the sun gear is small and the difference in diameters of the ring gears also is small. Although this calculation actually considers only those losses which occur at the gear teeth, the effects of other losses such as th due to bearing friction and oil churning are relatively s and can be accounted for by varying the value of f sl ly, based on experimental data. Tests of Curtiss speed ducers similar to that shown in *Figs.* 3 and 4 have sh that use of this formula with a value of $f = .011$ will g results consistent with those obtained by test. Effect bearing friction and oil churning are minimized in th units by using ball bearings at all important points and the use of a low-viscosity oil (AN-O-4) bath.

This method of calculation for the unit pictured w $f = .011$ gives an efficiency of 83.7 per cent for the in stage and 75.6 per cent for the output stage, resulting a calculated overall efficiency of 63.2 per cent, wh checks closely with the results of an experimental calib tion shown in *Fig.* 5. This power unit has a rated out

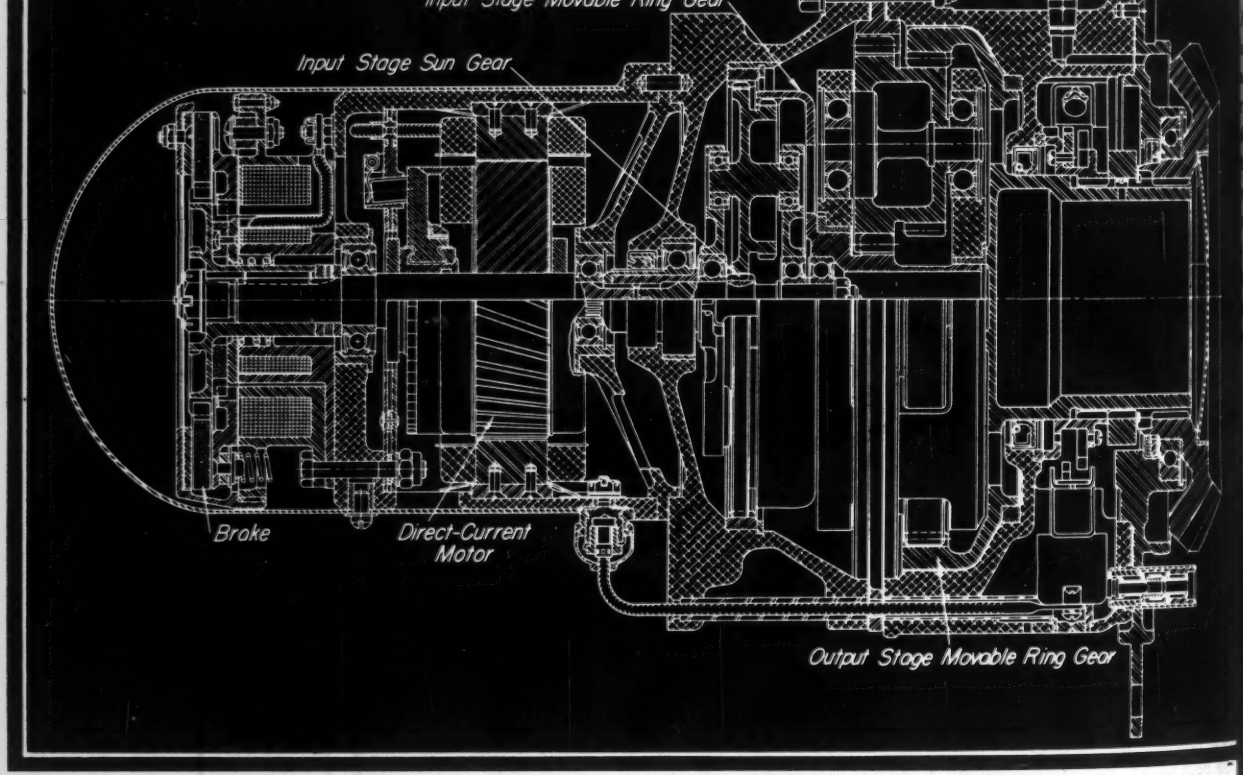
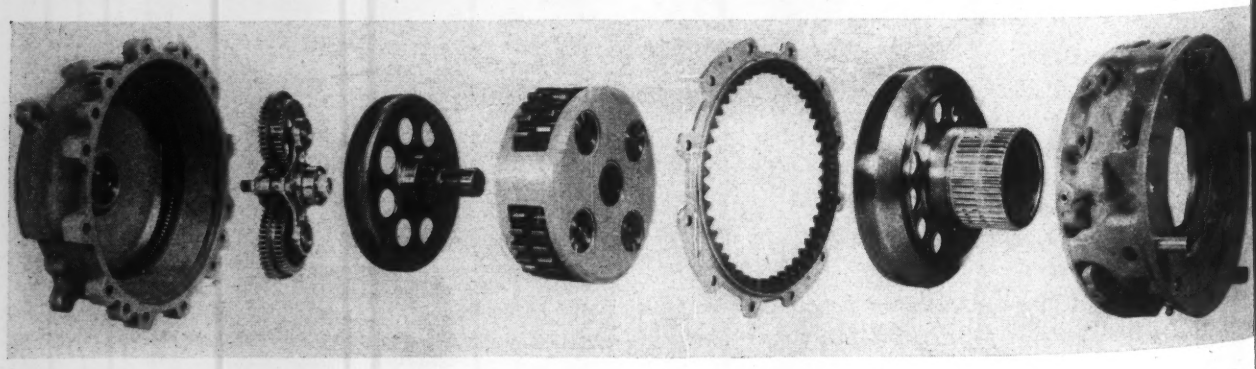


Fig. 3—Above—Cross section of a pitch-changing power unit rated at 75,000 inch-pounds output torque

Fig. 4—Below—Exploded view of speed reducer shown Fig. 3. Input stage is at left, output stage at right



ue of 75,000 inch-pounds and, complete with motor magnetic brake, weighs 57 pounds, while the speed ucer alone weighs 40 pounds.

When considering the design of a propeller speed-re- er unit, it is desirable first to consider the design op- ting condition. The required speed reducer rating is sen as that one which matches the total blade torque d developed at the maximum design speed of the pro- ler while operating at the angle of maximum twisting ment. Inasmuch as this angle of maximum twisting ment occurs at a relatively high blade angle—about degrees—which is attained only at high forward speeds high altitudes, at which time the engine is usually be- operated at a reduced speed, this rated load condition not an endurance condition. The endurance condition ually is that encountered at cruising revolutions per nute—about 77 per cent of the maximum—and a blade le slightly below the angle of maximum twisting mo- nt, resulting in a speed reducer torque load of about 50 cent of its rated output.

A further important point in the design condition is that rge part of the time the speed reducer simply holds the les in one position without being actuated. Actuation normal automatic constant-speed operation is in the re of a series of short jogs of a fraction of a degree tion at the output end of the speed reducer, one way then the other, at a frequency of about one energiza- per second. Thus, continued rotation is not a factor be considered in the design of a propeller speed re- er but provision must be made for the holding of high ue loads in one position under vibration for extended ods of time.

Designing for High Tooth Loads

On the basis of the design condition defined in the fore- ng, it has been found practical to use gear tooth loads about 10,000 pounds per inch of tooth face on 6.5 and diametral pitch gears in the output stage of the speed ucer. These tooth loads are made possible by careful ntion to the detail design of the gear teeth. The special r teeth used conform in general to the 20-degree stub th standards as far as addendum and dedendum are ncerned. However, a 25-degree pressure angle is used order to increase the beam strength of the teeth, and e dedendum of the planet gear teeth is deeper than ndard in order to incorporate a single full root radius. lso, the addendum of the sun gear is increased beyond ndard to increase the line of action or contact ratio.

Chordal thickness of the teeth is nonstandard since the hicknesses of mating teeth are made unequal in order to alance or equalize the bending stresses in mating teeth. his unequal thickness for equal stresses results, of course, om the inherently lower stresses of the buttress-shaped ternal gear tooth. Balancing of the stresses increases the verall strength of the train by increasing the strength of e external planet gear teeth. The resulting decrease in rength of the external sun gear teeth is not important ce these teeth carry relatively low stresses anyway.

Tooth thicknesses are adjusted to obtain equal stresses y laying out a first estimate of the mating teeth several times size and graphically determining the Y factors of both teeth by a modified Lewis method described in the following. Chordal thicknesses of the mating teeth are then adjusted on successive layouts until the Y factors found are equal. The original Lewis method of determining gear-tooth bending stresses assumes that the total load on the gear is applied to the tip of a single tooth whereas, in most gears in common use, all of the load is taken on a single tooth part of the time, and on two teeth during the remainder. On the driving gear, tooth contact begins at the bottom of the active depth and ends at the tip of the tooth. However, before the point of contact has reached the top of the tooth, the next following tooth has come into action and the load is then shared by two teeth. Thus the bending stress in any one tooth is a maximum when the next tooth comes into action.

In order to determine the true maximum bending stress, therefore, the modified method for determination of the Y factor of a tooth considers the tooth load as acting at that

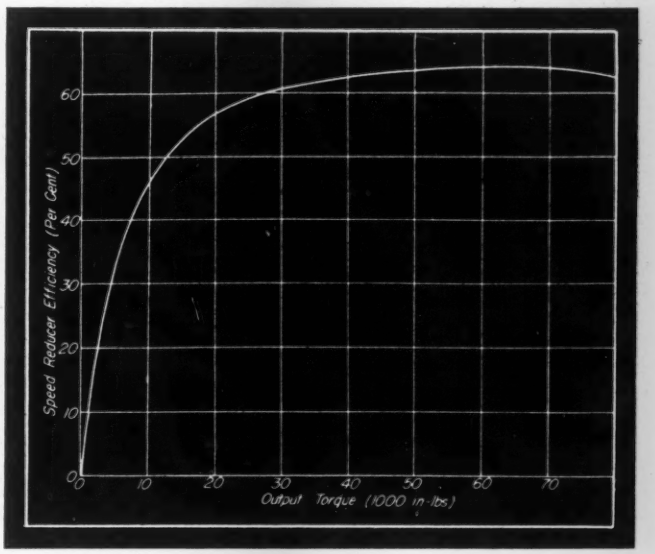


Fig. 5—Curve shows how speed-reducer efficiency varies with load. Reduction ratio is 7056:1

point on the leading face where there is contact at the instant when the next following tooth comes into action. This point can be accurately determined on the large-scale layout by constructing one tooth profile so as to pass through the intersection of the line of action and the addendum circle of the mating gear. Then the location of the tooth load corresponding to the maximum tooth bending stress will be at the intersection of the centerline of the next previous tooth and the line of action. Determination of the Y factor may then be made by conventional methods. The method illustrated in *Fig.* 6 consists of drawing a series of parallel lines, g, perpendicular to the centerline of the tooth and across half the section of the tooth. To find the "weakest" section, measure the lengths of these lines g_1, g_2, g_3, etc., and their perpendicular distances from the intersection of the tooth load and the vertical centerline, h_1, h_2, h_3, etc. Then if $x = g^2/h$, the values of g and h which give the smallest value of x indi-

cate the weakest section. Using this value of x:

$$Y=\frac{2x}{3CP}$$

where CP = circular pitch.

Maximum tensile stress in the tooth then is equal to the bending stress due to the tangential component of the tooth load, minus the compressive stress due to the radial component, times a root-radius stress concentration factor.

Tooth loads on the planet gears from the two ring gears are of course in opposite directions, forming a couple which tends to deflect the planet gears out of their normal position parallel to the axis of the ring gears. This couple must be restrained by the planet-gear bearings and in turn by the planet-gear cage. Such a loading on the cage has a tendency to deflect the two cage side plates angularly with respect to each other, i.e., to rotate one plate with respect to the other. It will be seen that such rotation of one plate with respect to the other will misalign the planet gear bearings and tend to cause end loading on the planet and ring gear teeth since the axes of the planet and ring gears do not remain parallel. For these reasons it is desirable to design the cage to be as rigid as possible compatible with other considerations. This condition becomes particularly important in the output stage where the tooth loads are highest and the distance between them is greatest. In the output stage of the unit shown in *Figs.* 3 and 4, maximum rigidity compatible with weight requirements is obtained by the use of a one-piece cast aluminum alloy cage.

In order to nullify the effects of such cage deflections as may be encountered, it is desirable to utilize types of bearings which can accommodate some misalignment. It is also possible to minimize the stress concentrating effects of end loading on the gear teeth by the use of "crowned" teeth. It is probable that crowning only of the planet gear teeth provides a satisfactory solution to this proble

Planet-gear cage deflection and the resulting misali ment of planet and ring gear centerlines also produce tendency of the planet gears to "walk" fore or aft out the ring gears, the direction depending upon the direct of rotation. This condition results in a thrust load appl by the cage to those bearings restraining it on either si The thrust load, of course, results in increased friction these restraining bearings which can become an import factor in the overall speed-reducer efficiency. This is p ticularly true of those bearings on the forward side of t output-stage cage which also support the input-stage p gear. Since the sun gear is a high-speed, low-torque me ber, any friction torque applied to it assumes increased i portance. Hence it is desirable, where possible, to isol

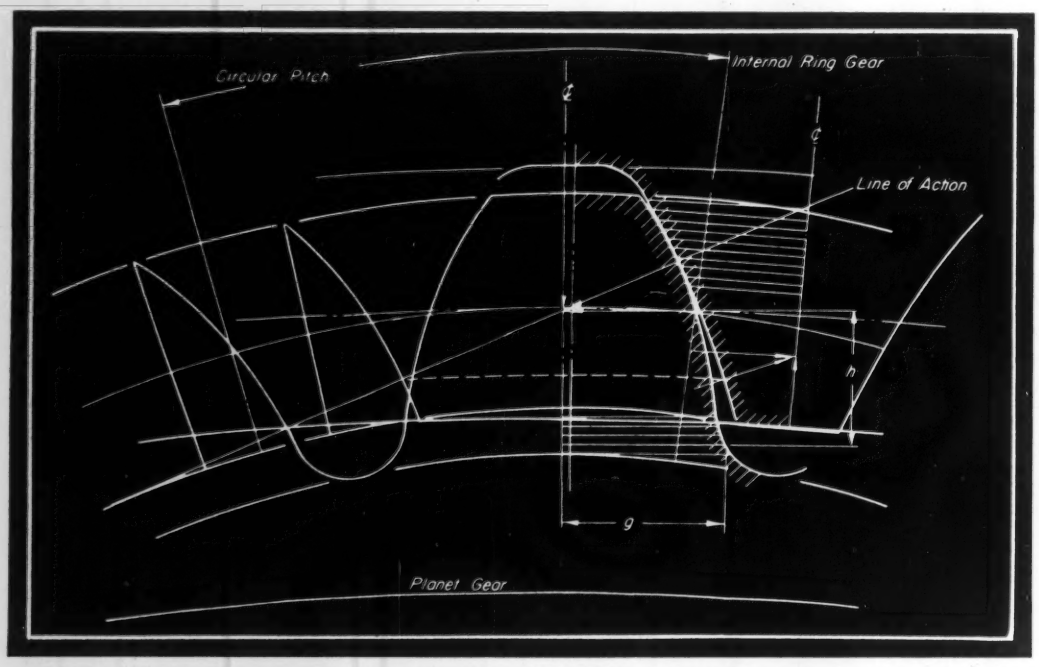
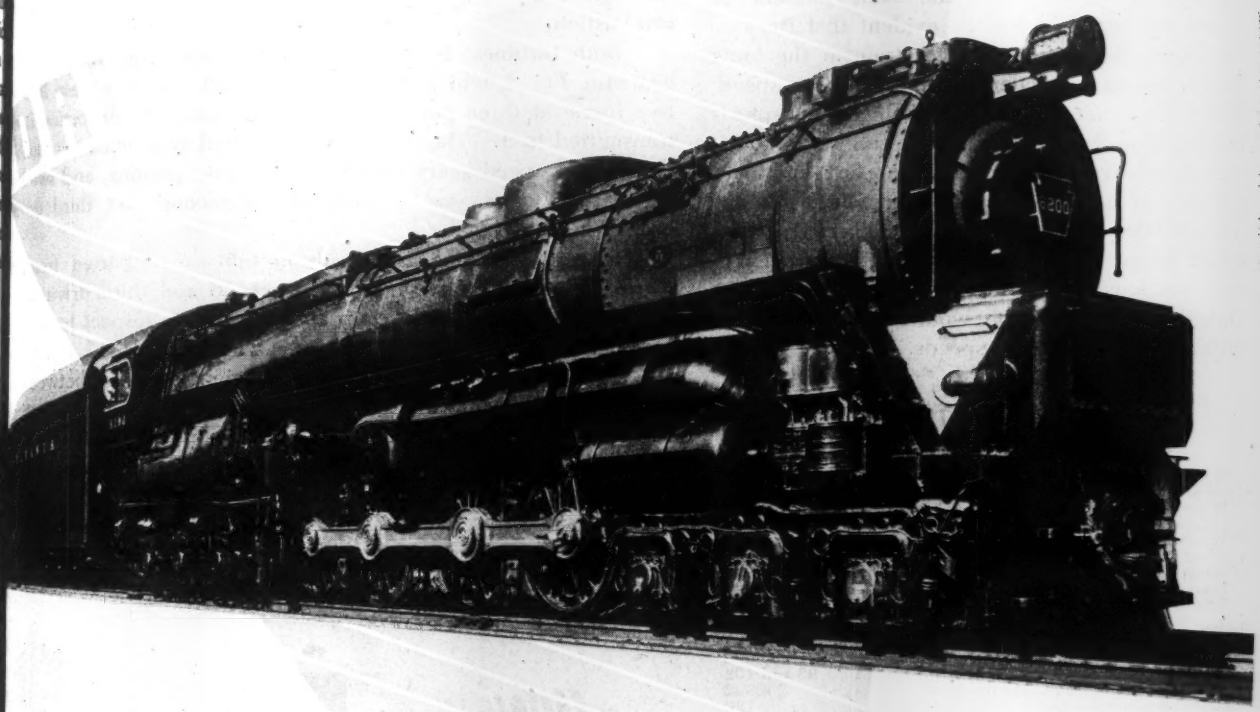


Fig. 6—This type gear-tooth layout used to determine a equalize bendi stresses in mating tee

any thrust of the output stage from the high-speed mem bers of the input stage.

It will be seen from *Fig.* 3 that the output-stage plane gear cage is not piloted or centered in the housings o ring gears by any bearing surfaces. Radial support fo the cage is provided by the teeth of the planet gears mesh ing with the ring gears. The cage carrying the four plane gears is thus permitted to "float" on the gear mesh points and to center itself automatically to divide the torque load as accurately as possible between the planet gears. This feature has been found to be desirable in order to avoid overloading of the teeth of one planet gear while others are only lightly loaded and to avoid friction losses on any cage bearings which try to restrict the normal centering ac tion of the planet gear cage.

With reasonable attention to detail design points, and particularly to results of the deflections of stressed parts, it is possible to produce a highly efficient, lightweight speed reducer based on the two-stage compound epicyclic prin-ciple. Under extreme temperature, vibration and load conditions such units have proved highly successful.



Steam Turbines Drive New Locomotive

By Charles Kerr Jr., J. S. Newton and W. A. Brecht
Westinghouse Electric & Mfg. Co.

ALTHOUGH new developments usually are first designed and built on a relatively small scale, the Pennsylvania Railroad company's new Class S-2 geared steam turbine locomotive, with a rating of 6900 shaft horsepower, is one of the most powerful high-speed locomotives in the world. Designed for fast, long-haul passenger and freight service, it can operate at speeds up to 100 miles per hour. Propulsion equipment was designed for the maximum rating considered practicable for a rigid-frame, four-driving-axle unit, operating with a conventional fire-tube boiler at pressures and temperature commonly used with such boilers.

As shown in the head illustration, the new Baldwin-built locomotive has a 6-8-6 wheel arrangement. The Westinghouse turbines and reduction gears drive the two center pairs of driving wheels, to which the other driving wheels are coupled by connecting rods. Leading particulars of the locomotive, which burns coal and is stoker-fired, are given in TABLE I.

Among the advantages offered by the turbine over the conventional drive are

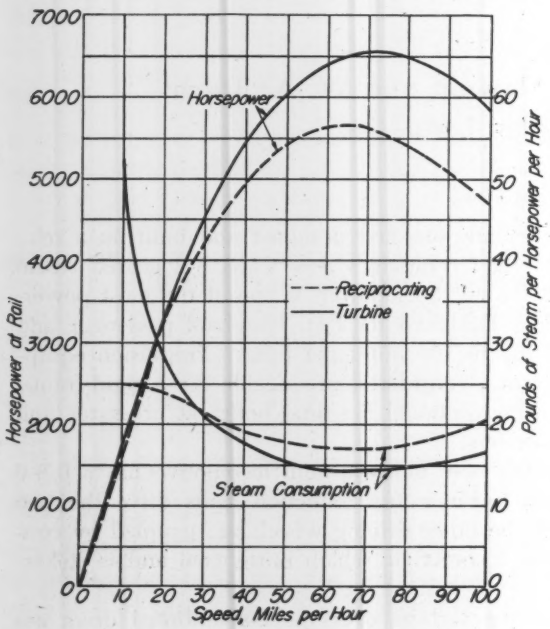
higher efficiency and smoother propulsion. *Fig.* 1 shows how horsepower and steam consumption compare with a conventional two-cylinder reciprocating steam locomotive having the same weight on the drivers and a boiler designed for equal evaporation. For the same amount of steam consumed, it will be evident that the turbine unit can handle a larger train at the same speed, or an equivalent train at higher speed. Although steam consumption is higher at speeds below 27 miles per hour with full steam flow, this is relatively unimportant for fast long-haul service. Actually, even with heavy trains it has seldom been necessary to use full throttle in this speed range. In a turbine drive intended specifically for lower-speed service the turbine blading and the gear reduction would be designed for higher efficiency at low speeds.

TABLE I

Particulars of Turbine Locomotive

General:	
Wheelbase, total engine, feet	53
Driving wheels, outside diameter, inches	68
Weight of engine plus tender, pounds	1,029,400
Weight of engine, pounds	580,000
Weight on drivers, pounds	260,000
Tractive effort at starting, forward, pounds	70,500
Tractive effort at starting, reverse, pounds	65,000
Boiler:	
Working pressure, pounds per square inch	310
Capacity, pounds of steam per hour	95,000
Diameter front and back, inches	93 and 102
Heating surface, total, square feet	4,992
Grate area, square feet	120
Superheating surface, square feet	2,050
Turbines:	
Steam pressure at inlet, pounds per square inch	285
Exhaust	Noncondensing
Maximum horsepower, forward	6,900
Maximum horsepower, reverse	1,500
Reduction Gears:	
High-speed ratio	5.16
Low-speed ratio	3.58
Total reduction	18.50

Fig. 1—Horsepower and steam consumption compared for locomotives with equal boiler capacity



Besides having a better steam rate, the Class S-2 locomotive also utilizes more efficiently the coal burned in the boiler. Due to the small wheels and short driving wheelbase it has been possible to equip this locomotive with a large boiler and a large grate. When developing full output, coal is burned at a relatively low rate and the boiler is easily drafted, resulting in more efficient combustion.

Both turbines, forward and reverse, are supported from the gearcase, *Fig.* 2, which in turn is supported from the main locomotive frame at three points in such a manner that distortion is not transmitted to it. The gear is of the nested type with two double helical high-speed gears, two low-speed spur pinions, and two low-speed spur gears, one on each of the second and third driving axles, *Fig.* 3.

Since the complete propulsion unit is supported from the spring-borne locomotive frame, the second and third driving axles must be permitted to move up and down with respect to the low-speed gears which surround them. The "cup drive" which permits this motion is, in effect, a misalignment coupling between the low-speed gear and the locomotive axle, and is housed entirely within the gear shown in *Fig.* 3. In addition to permitting up and down motion, the cup drives are torsionally flexible, thus protecting the gearing and turbines from shock loads.

Helical pinion and gears are hardened and ground, the tooth loading and contact hardness (450 brinell) of the high-speed pinion

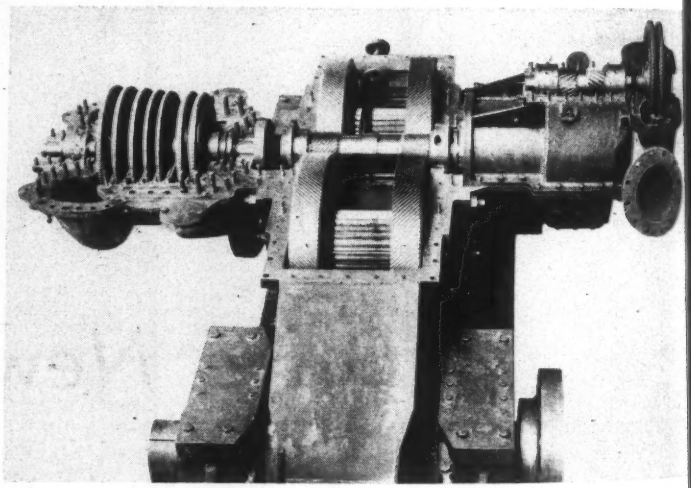
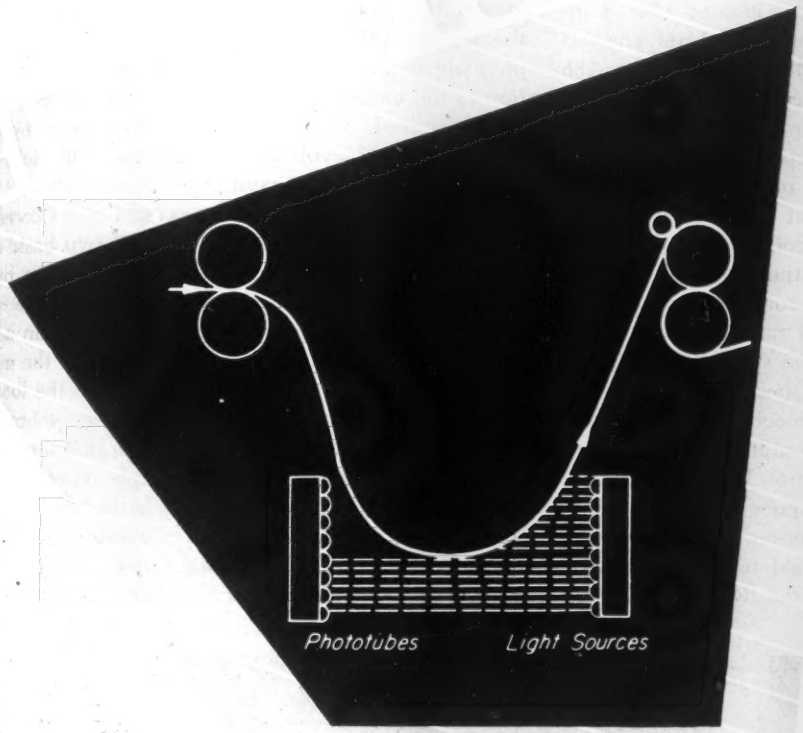


Fig. 2—Forward turbine, upper left, drives axles through double-reduction gearing. Reverse turbine drives high-speed main pinion through auxiliary reduction gear and hydraulically operated clutch

being more than double the values commonly used in this type of gear. The low-speed spur gears are hobbed, hardened to approximately 500 brinell, and ground. They are of the type used on main line electric locomotives.

As will be noted from *Fig.* 2, the forward turbine is of the impulse type, consisting of a Curtis stage followed by five full-admission Rateau stages. At 100 miles per hour locomotive speed the turbine rotates at approximately 9000 revolutions per minute. It is connected to the high-speed pinion at the opposite side by a quill shaft extending through the hollow pinion. Steam enters the turbine through four 3-inch pipes each of which is connected to a nozzle group covering approximately 25 per cent of the peripheral area of the Curtis-stage blading. Four cam-operated valves, one

(*Concluded on Page* 192)



Electronics Insures Accurate Loop Control

By L. U. C. Kelling
Industrial Control Engineering Division
General Electric Co.

ELECTRONICS, properly utilized, continues to play a role of ever-increasing importance in the control of machines. Among the most significant applications, from the standpoint of the machine designer, is the automatic loop control used on continuous-strip processing lines.

If the reader has had the opportunity of viewing such processing lines in operation, he will have noted that the strip material is permitted to hang in slack loops between various sections of the lines. These loops serve a definite purpose, some of the reasons for their use being: Elimination of tension between sections of lines, isolation of intermittent or pulsating feed in one section from another section, and accurate guidance of the strip into a subsequent machine, such as a side trimmer or a coiler. Then too, there are the large loops which provide storage between sections, permitting continuous operation of one section while another is stopped momentarily for splicing, removing coils, inspection, or adjustment.

It no doubt will be apparent that to maintain a loop properly the speed of the

material in one of the two sections adjacent to the loop must be controlled either manually or automatically. In most cases, as might be surmised, automatic loop control will do the job much better and faster than is possible by manual operations.

To give the reader a comprehensive idea of what is involved in loop maintenance, the following brief analysis of payoff-loop control is offered: Material is fed into the loop (see *Fig.* 1) from the payoff coil and is withdrawn from the loop by the side trimmer shear at a constant linear speed. Obviously, for the loop to ride steadily at a fixed position in space, the material must be fed from the coil at the same linear speed as that at which it is withdrawn from the loop. If the payoff coil rotates too rapidly, the material will accumulate and the loop will become deeper. Similarly, if the payoff coil rotates too slowly, the loop will become more shallow. As the coil diameter decreases it is necessary to increase the speed of the payoff motor to maintain the same linear speed of the material.

Shown in *Fig.* 2 is a simplified mechanical layout and elementary wiring diagram of a tin-mill shear line which incorporates two photoelectric thyratron loop controls. The steel is pulled through the side trimmer shear at a preset constant speed. Speed of the direct-current shunt motor which drives the payoff coil is varied to hold the payoff loop in position, and the speed of the flying shear motor is varied to maintain the position of the second loop. Wer manual operation employed, the operator would hold the loops by varying the speeds of the direct-current shu motors with field rheostats 11R and 13R. For automat operation, however, these field rheostats are set at the maximum resistance position. Then, in accordance wi the loop position relative to the phototube holder, ea photoelectric loop control supplies the proper field excit tion to the motor and so maintains the loop position. Th line is started, stopped, or set to any speed by changi the armature bus voltage of the motors with field potenti meter 10R of the generator.

Basic Functions of Automatic Loop Control: A automatic loop control must consist of two basic element First, there must be a means of detecting the position o the loop and, in some cases, the approximate speed re quired for the coil. Second, there must be an adjustable speed motor control to control the speed of the material i one of the sections of the line adjacent to the loop.

Automatic loop controls employ either photoelectric o mechanical means for determining the position of the loop A mechanical limit switch or a simple on-off photoelectri relay can be used to determine whether the loop is abov or below a definite position. A roll riding on the loop ma operate a rheostat or the plunger of a solenoid reactor an so indicate the position of the loop. Then too, the ap

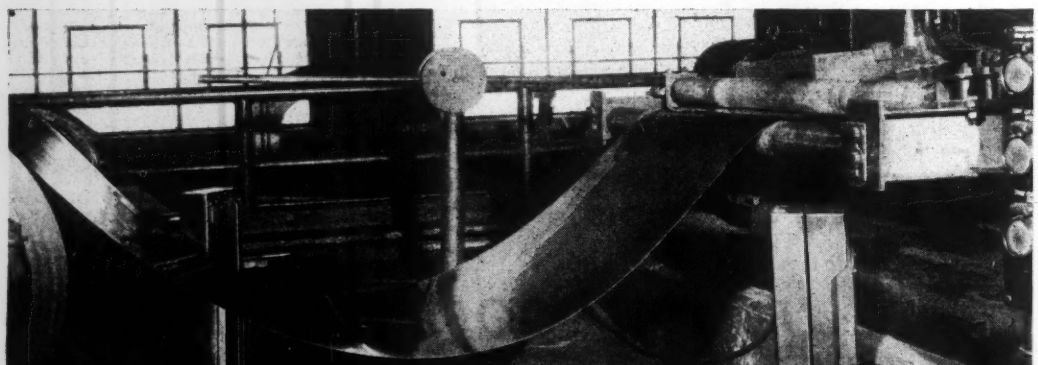
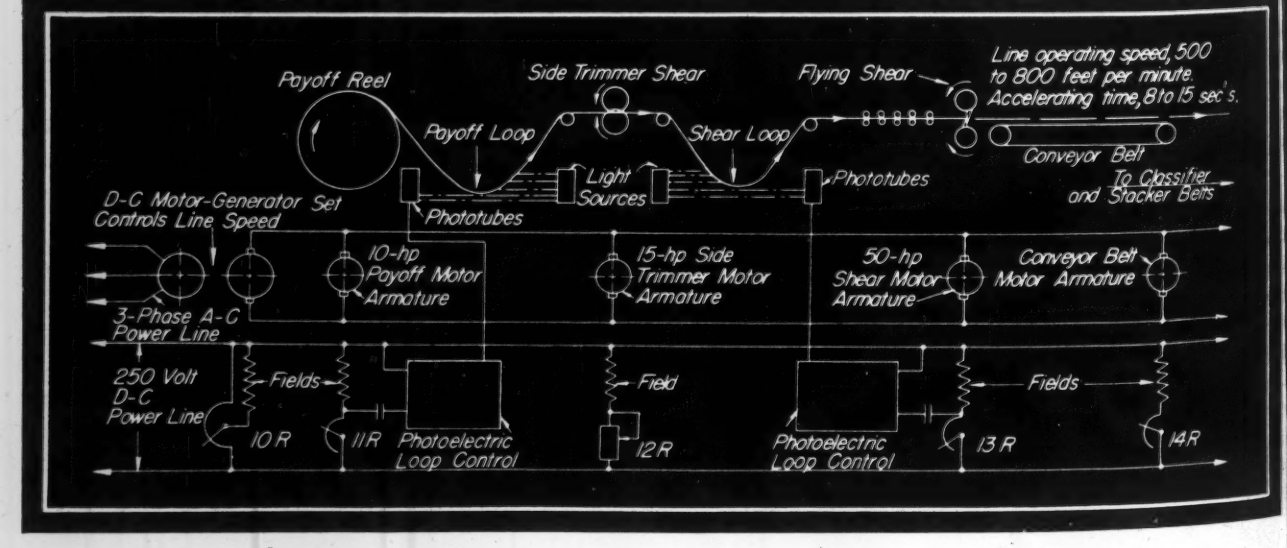


Fig. 1—Left—Diagon positioning of ligh source and phototub holder for loop contro

Fig. 2 — Below — Tin mill shear line in corporating two photo electric thyratron loop controls



ximate speed required of the coil be indicated by a rider-roll rheoat operated by an arm riding on the face of the payoff coil which slowdecreases in diameter. Going furr with the photoelectric means, act position of the loop within a finite control zone may be detected having the loop intercept a curtain light passing between a light source d a phototube unit.

As for the adjustable-speed motor ntrol, this may consist of a simple gging relay or contactor which inreases or decreases the speed slightfrom some preset value. Again, it ay consist of a rheostat which conols the speed of the motor. Conuously adjustable electronic motor ntrol may consist of a thyratron be rectifier circuit, an electronically ntrolled motor-generator set, or a gh-vacuum tube connected in series parallel with the field.

TYPES OF AUTOMATIC LOOP CONOL: The aforementioned means of tecting loop position and controlling otor speed may be combined to rm many types of automatic loop ntrol that differ widely in their aracteristics. To avoid making the scription of each of the six repretative types discussed in the folwing too general, and therefore ague, pertinent assumptions are ade regarding the application of ch control, to wit: Each shall be sed as a field current control of a rect-current shunt motor; the armare voltage of this motor shall be furshed by a direct-current motor-genrator set which controls the speed of e line; the material shall be drawn t of the loop at a constant linear eed; the loop control shall regulate e speed of the material feeding into e loop, and the material shall be fed om a pinch roll stand or direct from the coil of mateal which is constantly decreasing in diameter.

TYPE A: A loop-rheostat loop control employs a roller at rides on and is supported by the loop and operates a heostat connected in the field of the motor. Thus, if the op is too low, the field of the motor is strengthened, the otor slows down, the material is fed into the loop at a ower speed, causing the loop to rise. If the loop is too igh, the same sequence of events will occur but of course the opposite direction so as to lower the loop. In an proved version of this type of control, the roller is conected to the armature of a solenoid reactor which conols the output voltage of a thyratron rectifier circuit supying field current to the motor.

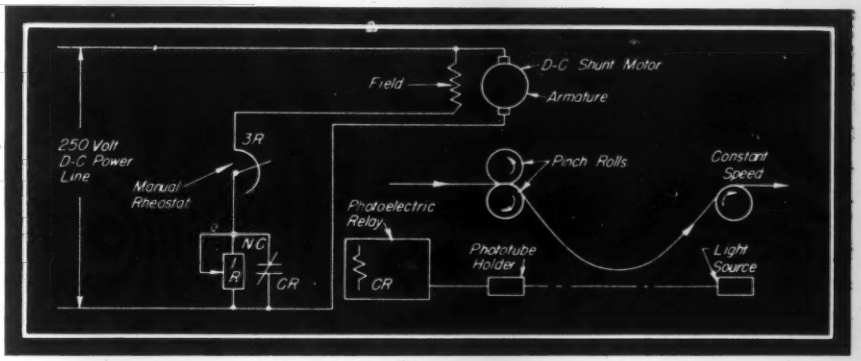


Fig. 3—Above—One-point loop control with manual rheostat

Fig. 4—Below—Two-point loop control similar to system of Fig. 3

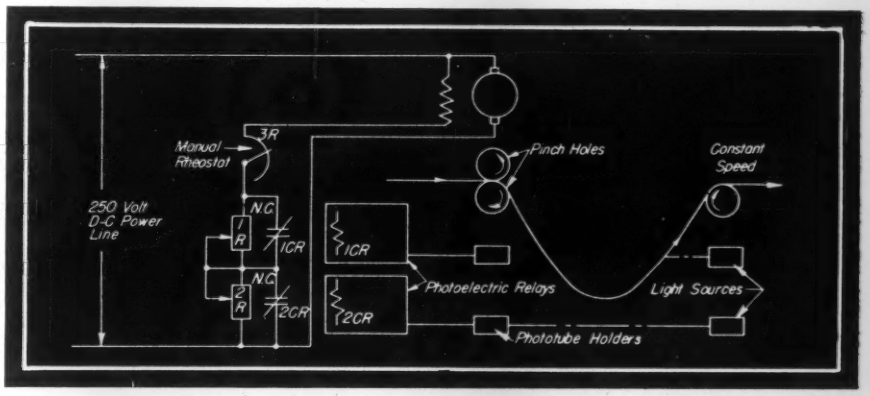
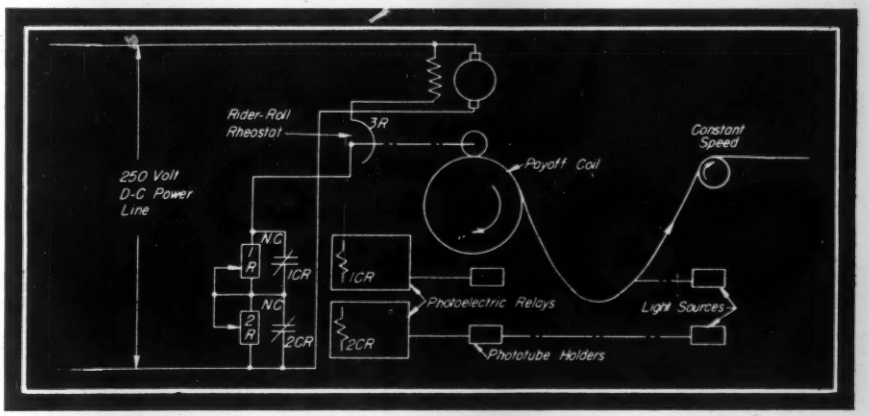


Fig. 5—Below—Combination of two-point control and rider-roll rheostat



TYPE B: Shown in *Fig.* 3 is a one-point loop control consisting of a manual field rheostat and a small resistor connected in series with the motor field. The contacts of a simple photoelectric relay are connected across the small resistor, and the photoelectric relay is controlled by a beam of light passing under the bottom of the loop. In operation, the manual rheostat sets a material speed slightly higher than that at which the material is withdrawn from the loop. Then, when the loop drops and the light beam is intercepted, the photoelectric relay contacts close and produce a second slightly lower speed which is less than that at which the material is withdrawn from the loop. Because these two speeds bracket the desired speed, the control will shift continuously from one speed to the other

in order to maintain the required loop position.

TYPE C: The two-point loop control shown in *Fig.* 4 is the same as the one-point control, type *B*, except that two photoelectric relays and two small field resistors are used. The photoelectric relay with the light beam in the high position of the loop is connected so that the speed of the material will increase when the loop rises above this beam. The photoelectric relay with the light beam in the low position is connected so that the speed of the material will decrease when the loop drops below that beam. With the manual rheostat set as close to the proper speed as possible, the two photoelectric relays will correct for any accumulated differences. Change in speed caused by each operation of the photoelectric relays should be great enough to cause the loop to be moved well away from the light beam position toward the center of the region between the two light beams. This, combined with close manual setting of speed, greatly reduces the control operating frequency

TYPE D: The loop control illustrated in *Fig.* 5 is a combination of a rider-roll rheostat and the two-point control. It is used on payoff loops where the material is fed directly into the loop from the coil of material, thereby requiring that the speed of the coil increase as the diameter decreases. The rider-roll rheostat replaces the manual rheostat of the basic two-point control. The movable contact of the rider-roll rheostat is fixed to an arm riding on the coil surface and the rheostat is designed to set coil speeds which are approximately correct for all diameters of the coil. Any accumulated differences are corrected by the two-point control. It is also possible to use one-point control with a rider-roll rheostat.

TYPE E: The motor-operated rheostat loop shown in *Fig.* 6 is a combination of a motor-operated rheostat and

(*Continued on Page* 172)

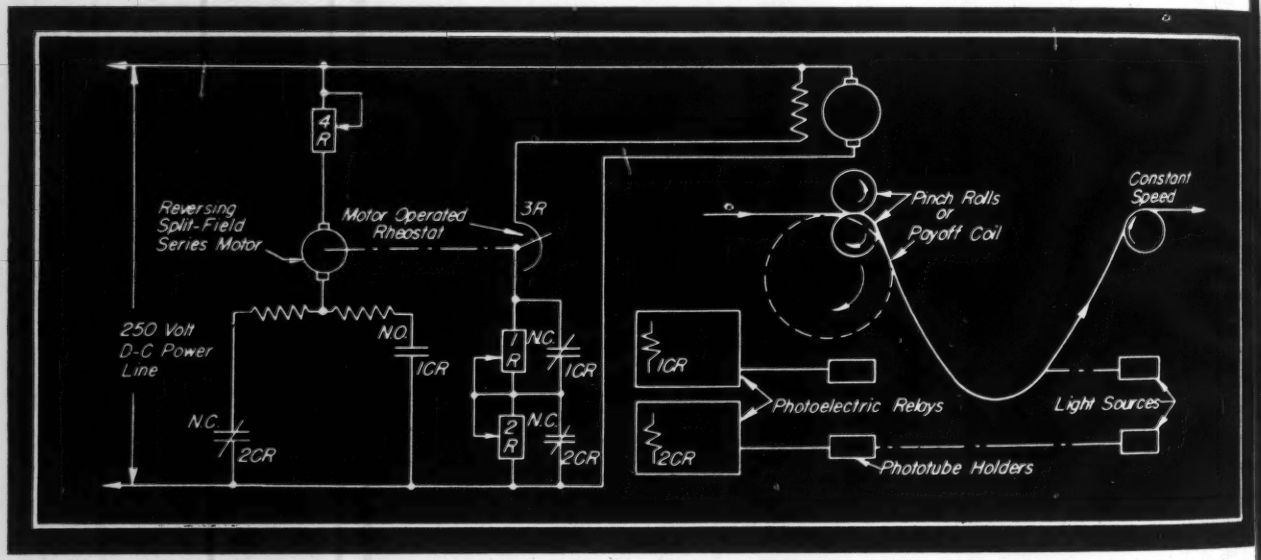
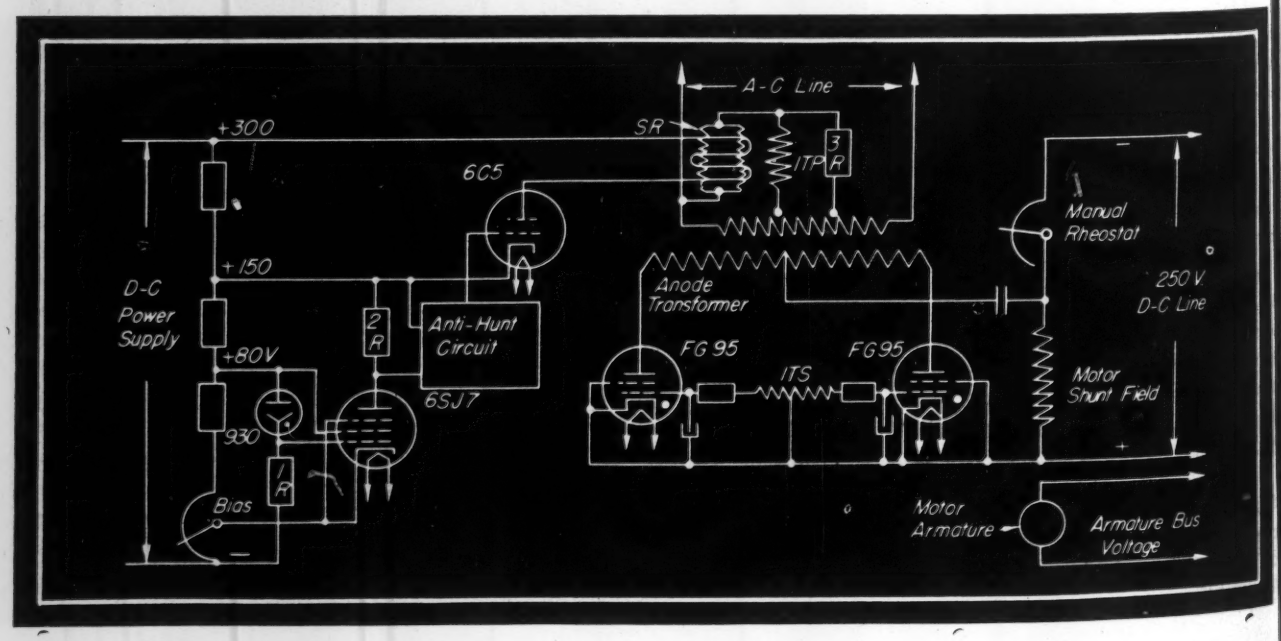
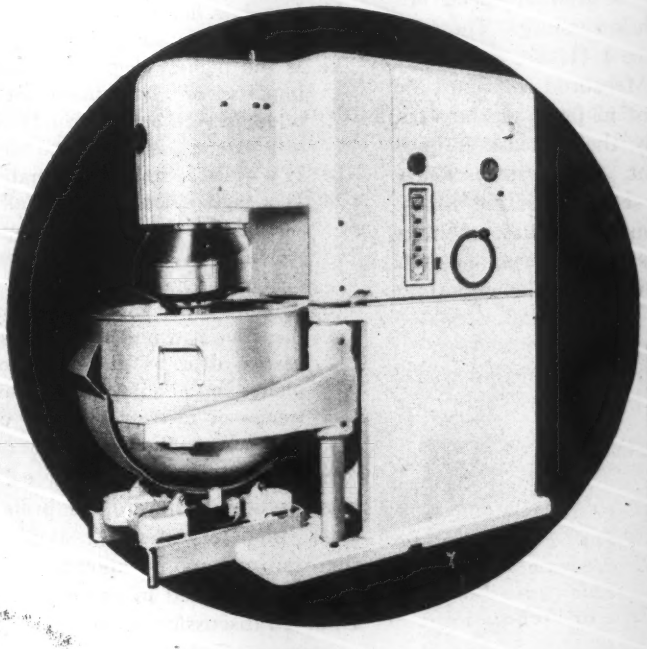


Fig. 6—Above—This system combines the two-point control (Type B) with motor-operated rheostat

Fig. 7—Below—Typical all-electronic control consists of detection, amplification and rectification circuits





Selecting Drives for Speed Control

By Ernst L. Schwarz—Kast
Armour Research Foundation

Part I—Mechanical Drives

MANY designers of machinery are vitally interested in a reliable, variable-speed drive for application to machine tools, printing machinery, textile machines, testing machines and other equipment such as the processing machine in the head illustration which has a Reeves variable-speed transmission built into the head.

Today there is a demand for increased output of machine tools to the limit which requires operation at the highest possible speed. This optimum speed is determined by the type of cutting tool, the material to be machined and the kind of work to be done. The speed might change in a wide range with these factors. There are also many phases during the operation of these machines which require a temporary slow-down, such as cross-feeding of a tool, slow-down of a milling cutter for sharp corners, accurate positioning of tool to work and the like. In general, there is certainly no doubt about the economical advantage of a speed control but it seems worthwhile to have more accurate data about the actual savings which may be obtained by the selection of a more adequate variable-speed device, as may be seen from the following:

Comparative tests were made with two identical lathes, one equipped with a variable-speed drive, the other with a gear shift (1)*. In both cases the drives were arranged in the base below the headstock and the spindles were driven by belt. Total range of speed variation was roughly the same in both cases, the only difference being that the variable-speed drive permitted a gradual speed adjustment and also an adjustment while running. The gear shift had only definite steps, sequence 1.41, and the speeds had to be preset before starting. Measured average time saving was at least twenty per cent in favor of the variable-speed drive, and consequently the increase in production was more than 25 per cent. This shows clearly the important influence which the selection of the appropriate speed control has on the output of the machine.

There are innumerable variable-speed devices and it is impossible to discuss them all or even only the important

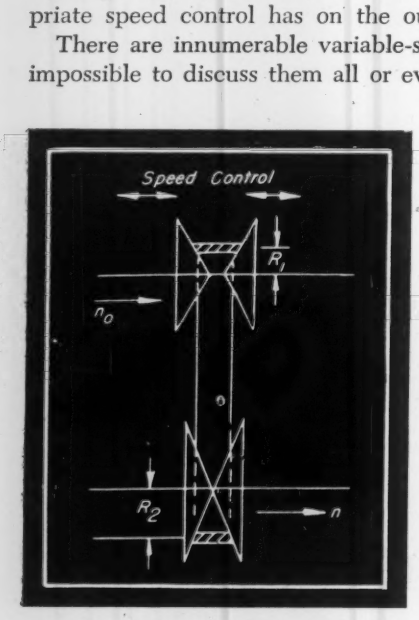


Fig. 1 — Schematic diagram for a variable-speed V-belt or V-chain drive

ones in this series of articles. All the devices, however, may be classified into three groups: Mechanical, hydraulic and electric.

An attempt will be made to review these groups and critically compare their main characteristics. Thus these articles will be a guide for those designers, specialists in other fields, who are only occasionally and infrequently engaged with problems of speed control.

For the selection of the most appropriate system, many factors have to be known. Some are special and only related to the individual assignment. These cannot be covered in this series, but some are of general importance and they will be discussed. The following check-list contains important considerations when planning a drive.

1. If speed variation is necessary, what is its range?
2. Is it desirable to have a gradual variation, an infinite number of steps, or a few single definite steps?
3. Does the speed require adjustment while running or can it be preset before starting?
4. How does the torque requirement change with the adjusted speed?
5. Does the load remain constant at any adjusted speed?
6. If the load changes, should adjusted running speed remain constant or may it vary with the load, and what are the permissible deflections?
7. What changes of the load have to be expected at the different speeds? Is there any possibility that the load will become zero or even negative?
8. In case of an overhauling load, should the speed drive act as a regenerating brake?
9. Can the machine run both ways?
10. Is an extremely quick stop or reversal essential?
11. Is a low first cost imperative or is it more desirable to have the operating costs low, even at higher first cost?
12. Is the highest degree of safety in operation of decisive importance? Is it important to have a minimum wear under severe service conditions? Or should the necessary care and maintenance be a minimum?
13. Is a quiet or noiseless operation required?
14. Is a high efficiency essential? Is the amount of power wasted by the respective speed reduction method objectionable? Or is the dissipation of generated heat difficult or undesirable?
15. Are there any unusual surrounding conditions to which the drive is exposed? For instance, moisture, vapors, excessive dust or dirt, acids or fumes, explosive atmospheres, installation in open air, exposure to weather extremes or excessive heat or cold? Is the drive exposed to shock or vibration?

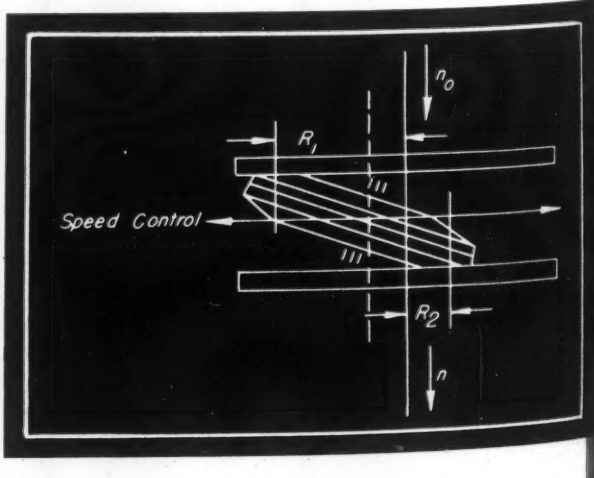
Simplest forms of mechanical devices for speed variation are step pulleys, cone pulleys, sliding gear transmissions, planetary multiple-speed gears and the like. All these are commonly known and thorough information on them is contained in every handbook on machine design. Detailed discussion in this article, therefore, does not seem necessary. These simple devices are inexpensive and reliable and sometimes convenient, especially for small capacities and where only some definite speed steps are sufficient.

Besides these simple mechanical devices there exists a great number of mechanical devices for gradual speed variation. With few exceptions they transmit power by friction and the majority can be classified into the following groups:

1. V-belt, V-chain or V-ring with conic pulleys
2. Double friction cone between two plane disks
3. Two friction disks
4. Planetary.

V-belt or V-chain system with conic pulleys and changeable pitch diameter is shown schematically in *Fig.* 1.

Fig. 2—Below—Double friction cone drive for speed control



*References in parentheses are listed at end of article.

e speed ratio R_1/R_2 is not more than 4:1 the pitch of e of the pulleys may be fixed, while the pitch of the her and the center distance between both pulleys are justable. A countershaft can be arranged with one adstable V-belt drive between input shaft and a counterft, and another between the countershaft and output ft, giving a total ratio up to about 16:1. In another pe of unit, speed range up to 16:1 is obtainable by varyg the pitch of both pulleys, with the center distance ed.

For the transmission of power usually a special V-belt is ed. Some devices use a metal ring as a friction element, me a special chain.

Double friction cone system between two plane disks, g. 2, uses one disk connected to the input and the other the output shaft. The power is transmitted by a double ne arranged between the disks, these being pressed ainst the double cone to provide the necessary friction. he cone is supported by antifriction bearings and is mov-

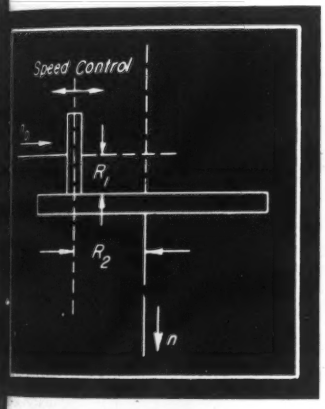


Fig. 3—Left—Basic diagram for a variable - speed drive utilizing two friction disks as shown

le to or from the center of the plane disks, giving a eed ratio R_1/R_2. The friction surfaces of the disks or f the cones or both are lined with a friction compound nd the necessary normal force is obtained by pressing the isks together by means of a spring or automatically, deending upon the adjusted magnitude of R_1 or the input rque.

Two friction disk arrangement, *Fig.* 3, has surfaces of oth disks lined with a friction compound. Speed variaon is obtained by moving the vertical disk to or from the enter of the horizontal disk.

Some of the later developments use basically a planetary ar, two being shown in *Figs.* 4 and 5. The system in g. 4 utilizes two or three conical rollers *A* which are inined at an angle equal to their taper so that their outer ces are parallel to the central axis and supported in a rrier *B* coupled to the driving shaft. The rollers with eir pinions *F* represent the planet gears and mesh with n internal gear *G* attached to the power output shaft. The nrotating member is a traction ring *C*, the position of hich ring can be varied axially by means of a rack and ion. Contact between the rollers and the ring causes e rollers to rotate around their own axes and in turn to ve the output shaft.

Maximum output speed is obtained with the ring at the reme large end of the tapered roller. This system (3) offers full torque over the entire speed range.

The other planetary drive, *Fig.* 5, has its driving shaft connected to the sungear *A* of a planetary gear. If the driving shaft rotates, the planet gears *B* will rotate and transmit rotation to the internal gear *C* as well. By holding the internal gear the planet gears will travel around the sun gear and cause rotation of the driven shaft, which is connected to the planet gears. If the internal gear is driven or permitted to rotate at various speeds the speed of the driven shaft would change and even cause the driven shaft to rotate in the reverse direction. This variable speed for the internal gear is obtained from the constant-speed driving shaft through the spiral gear *D*, an adjustable V-belt drive and a worm gear and worm shaft *E*. An advantage of this system is that most of the transmitted power passes directly through the gearing from the driving to the driven shaft and only a small part through the variable V-belt drive.

Applications of some of the foregoing mechanical devices were discussed in an article published previously in

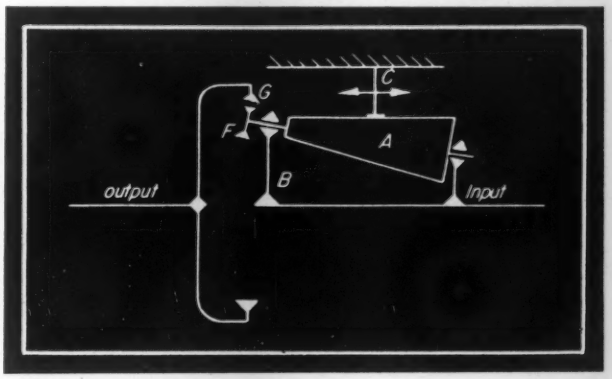
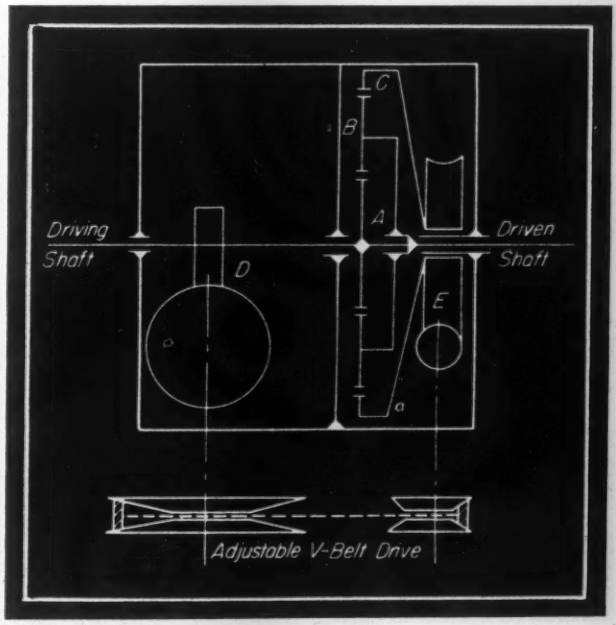


Fig. 4—Above—Planetary type of variable-speed unit. Speed control is obtained through conical rollers

Fig. 5—Below—Speed control is derived from an auxiliary V-belt in this planetary system



this magazine (4).

The maximum torque transmissible, the torque rating and the slip are dependent primarily upon the coefficient of friction between the contact surfaces and the magnitude of the normal force acting on these surfaces. The coefficient of friction is determined by the lining materials used and the normal force by the design of the device. Magnitude of the specific pressure necessary at the contact surfaces for transmission of the rated torque gives an indication of the quality, reliability and life of the device.

Each single apparatus can transmit a maximum torque under which slip and temperature rise at the friction surface remain within reasonable limits. This maximum torque may be called limit torque and the related po output the limit horsepower. This limit torque is no constant over the entire speed range, but decreases general with increasing speed for the reason given in following:

Assuming the normal force acting on the friction s faces is constant and the ultimate strength of the po transmitting means is also constant, it is clear that higher speed a greater part of the ultimate strength consumed to overcome the greater centrifugal forces a therefore less remains usable for the actual power tra mission.

It can be assumed that

$$T_{max} = T_{min}\sqrt{\frac{n_{max}}{n_{min}}}$$

and that the output horsepower rises with the speed the same proportion:

$$HP_{max} = HP_{min}\sqrt{\frac{n_{max}}{n_{min}}}$$

where T_{max} = limit output torque at lowest speed n_{min}, T_{min} = limit output torque at highest speed n_{max}, HP_{max} = limit horsepower output at highest speed n_{max}, and HP_{min} = limit horsepower output at lowest speed n_{min}.

In *Fig.* 6 is shown the limit torque and in *Fig.* 7 the lim horsepower at different speeds. *Fig.* 7 indicates that t limit horsepower output of a friction drive of this kind very different at minimum and maximum speed and t fact is important for the proper motor rating. The follo ing two cases are possible:

1. For a constant-horsepower drive, the motor capacity based on the horsepower capacity at lowest speed (po A, *Fig.* 7). At higher output speeds the capacity of t mechanical part of the drive would exceed by far t motor rating.
2. For a constant-torque drive, the motor is rated accordi to point *B* in *Fig.* 7, the horsepower capacity at high speed. There exists a definite danger of overloading damaging the mechanical part when the drive at low speeds would be loaded up to the full capacity of t motor. At lower speeds the motor horsepower rema constant but the maximum torque and the maximu horsepower transmissible by the drive decreases.

Mechanical devices offer advantages over electric a hydraulic systems under the following circumstances:

They are simple and compact insofar as constant-spe devices are concerned. They are appropriate whe only a mechanical transmission and no individual electr drive can be provided, or when operated in difficult su roundings, high temperatures, high humidity, moistur excessive dust, explosive atmospheres, and acid vapo They also have advantages when the drive is exposed shocks or severe vibration.

Following articles in this series will discuss princip underlying hydraulic and electric variable-speed drive

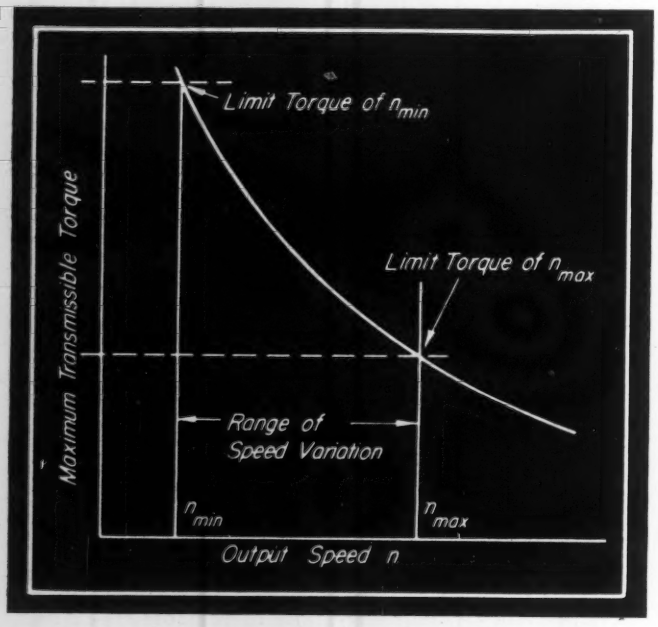
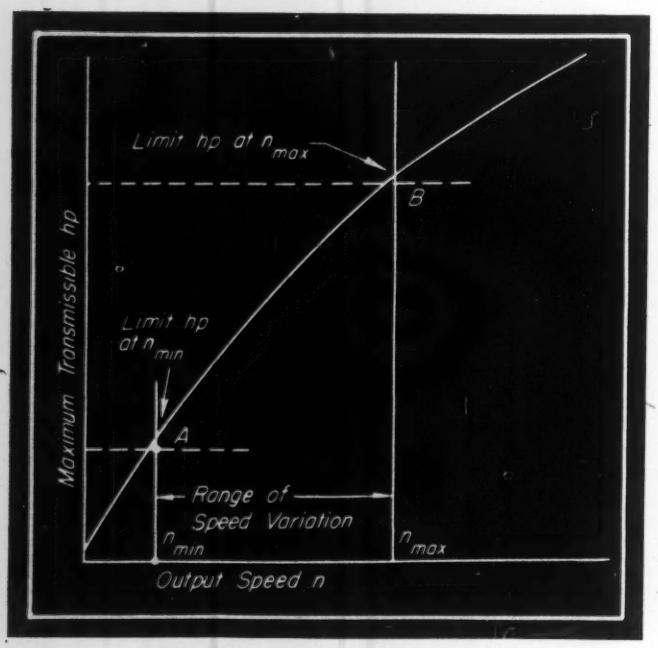


Fig. 6—Above—Torque capacity for speed range of mechanical transmissions

Fig. 7—Below—Horsepower capacity of mechanical transmissions for range of speed variation



REFERENCES

1. "The Advantages of a Hydraulic-Driven Lathe"—Germar, *Werkst Technik*, 1937.
2. "American Standard Definitions of Electrical Terms", 1941 Editi
3. *Kent's Mechanical Engineers' Handbook*, Eleventh Edition, Page 24-
4. "Specifying Variable-Speed Drives"—C. Carmichael, MACH DESIGN, April, 1944.

MACHINE Editorial DESIGN

Look to the Airplane!

DEVELOPMENT of aircraft since the early days of the war stands out in the achievements of engineers. In no other field has design progressed more rapidly since the idea was first put forward that this country build 50,000 planes for defense purposes—a suggestion which at that time was considered by many as merely "wishful thinking". Almost twice that number are now being turned out each year!

Much of the credit for this colossal production goes to those aircraft designers who displayed the necessary initiative in adapting designs to production-line technique. Equal or even greater credit must be given the designers who have continuously added refinement to refinement in improving the performance of all types of planes, finally winding up—insofar as bombers are concerned—with the Superfortresses now playing havoc with Japan.

That the development of this long-range bomber has been a stupendous undertaking is clearly emphasized from consideration of some of the details recently issued by the Air Technical Service Command. Weighing roughly sixty tons, the Superfortress has a cruising range of 3600 miles, a speed in excess of 300 miles per hour, and incorporates in its design more than 150 electric motors for the operation of its abundant auxiliary drives and controls. It reached its present stage of perfection only through having been under constant development for practically five years.

There is small wonder, considering the records that are being established by this plane and others, that designers in other fields are looking to the aircraft industry for ideas that might be utilized in developing their own designs. Future machines of all types unquestionably will benefit from the experience gained as a result of aircraft development. Reliability, reduction in weight, ease and automaticity of operation, split-second control, and problems of rigorous stress analysis are but a few of the factors of immediate as well as lasting influence on the design of other types of machines.

L. E. Jerry

P-47 "Thunderbolt" — Fighter-bomber *(Above)*. Pratt & Whitney Double-Wasp 2000-hp, 18-cylinder engine; speed, 450 mph; ceiling, over 40,000 ft.; four .50-caliber machine guns in each wing and 500 rounds of ammunition for each gun.

America's Armada of the Skies

Pacing certain victory for America and her allies, the aircraft pictured on these pages are representative of the mighty air fleet now lashing the common foe into unconditional surrender

P-63 "Kingcobra"—Fighter *(Right)*. New two-stage Allison 1500-hp V-12 engine; one 37-mm cannon through propeller hub and four .50-caliber machine guns; speed, close to 400 mph; service ceiling, about 35,000 ft.

SB2C-4 "Helldiver"—Dive bomber *(Below)*. Wright Cyclone engine; Curtiss electric four-bladed propeller; 20-mm wing cannon; eight 5-inch rockets under wing; wing racks carry 1000-lb. bomb load; internal bomb capacity, over 1000 lbs.; perforated-flap dive brakes.

"Airacomet"—Fighter *(Above)*. Propelled two thermal jet engines; speed, well over 400 tricycle landing gear, laminar-flow wing; over 5 tons; armament includes 37-mm and .50-caliber machine guns.

P-38L "Lightning"—Fighter-bomber *(Above)*. Two Allison V-1710-F30, liquid-cooled, in-line, 3000-hp engines; two turbosuperchargers; Curtiss constant-speed, full-feathering, three-bladed propellers; speed, over 425 mph; bomb load, 4000 lbs.; service ceiling, over 40,000 ft.; range, 1700 to 1800 miles.

XP-55 "Ascender"—Tail-first fighter *(Left)*. Allison 1275-hp engine; Curtiss electric three-bladed, constant-speed propeller with "kickoff" control; guns clustered in nose of plane.

P-51 "Mustang"—Fighter *(Left)*. Packard-built Rolls-Royce V-1650 engine, 1500-hp, with 2-stage, 2-speed supercharger. Curtiss electric four-bladed propeller; speed, over 450 mph; ceiling, over 40,000 ft.; bomb load, 1000 lbs.; four .50-caliber guns in wings.

P-61 "Black Widow"—Night fighter *(Right)*. Two Pratt & Whitney two-stage, two-speed supercharged 2000-hp engines; Curtiss electric four-bladed propellers; four .50-caliber machine guns in remote-control turret; four 20-mm cannon; equipped with radar detection equipment.

A-26 "Invader" — All-purpose bomber *(Below)*. Twin 2000-hp R-2800 series engines; Hamilton hydromatic propellers; remote-control gun turrets; NACA low-drag (laminar flow) airfoil wing section.

B-24 "Liberator" — [illegible] Pratt & Whitney [illegible] gines with turbo[illegible] service ceiling, ove[illegible] ten or more .50-cal[illegible] bullet-proof glass.

B-29 "Superfortress" — Heavy bomber *(Below)*. One and one-half times size of B-17; four Wright Cyclone 18-cylinder, radial, air-cooled R-3350, 2200-hp engines; Hamilton standard four-bladed propellers turn at [illegible] speed of engines; tricycle landing gear; remote-control turrets; web-type wing construction.

R-6 Helicopter—*(Below)*. Powered by 245-hp Franklin air-cooled engine; two-place model; has variable-pitch rotor blades.

[illegible] "Constellation"—Transport (*Right*). Four Wright R-3350, [illegible]00-hp, air-cooled radial, 18-cylinder engines; Hamilton standard hydromatic three-bladed propellers; speed, over 325 mph; service ceiling, 25,000 ft.; max. nonstop range, over 5000 miles.

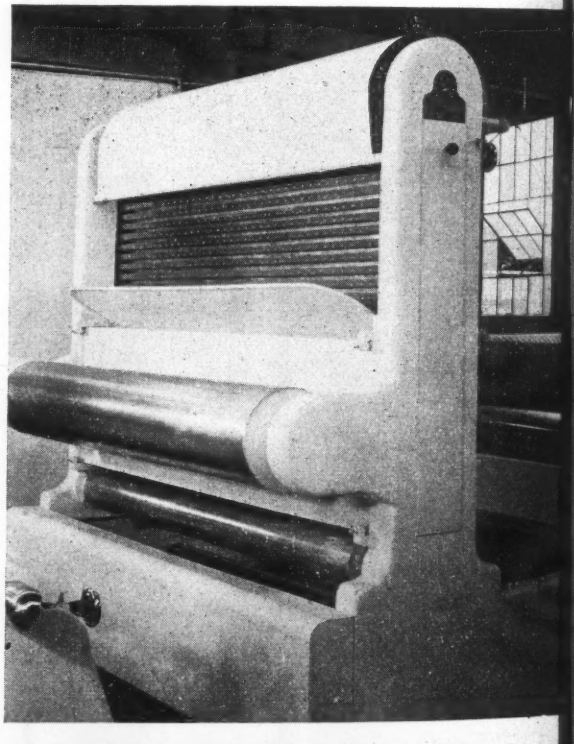
CW-20E "Commando"—Transport (*Left*). Two Wright Cyclone C18-B2, 2100-hp engines; max. cruising speed at 10,000 ft., 242 mph; useful load, 15,900 lbs.; Curtiss electric three-blade, quick-feathering, constant-speed propellers.

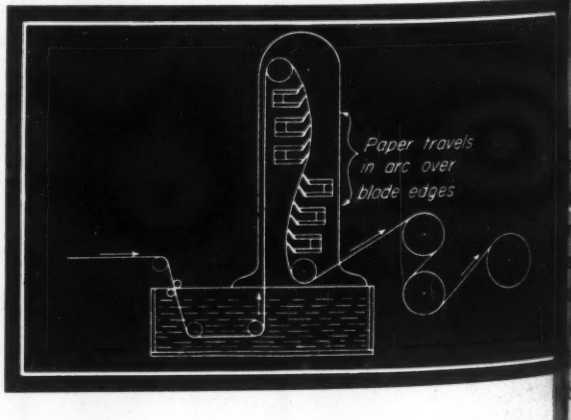
PV-1 "Ventura"—Bomber and subbuster (*Right*). Two Pratt & Whitney Wasp 2000-hp engines; speed, over 300 mph; service ceiling, over 25,000 ft.; Hamilton hydromatic three-bladed propellers; gross weight, 22,500 lbs.

Scanning THE FIELD for Ideas

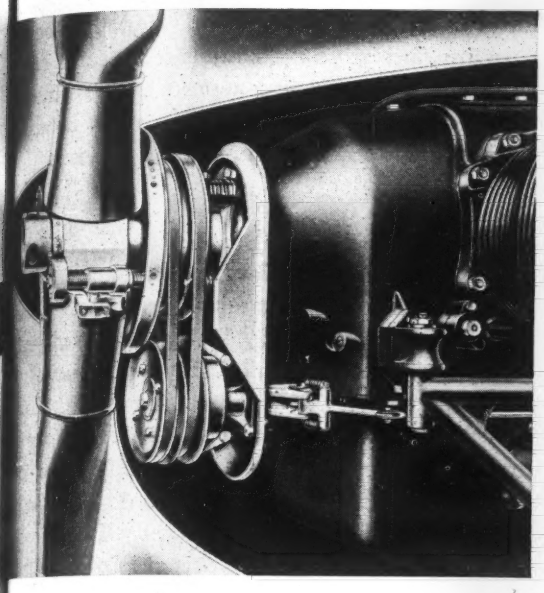
TRAPPED air facilitates the drying of wax paper sheet in the water finish machine, right, designed by Shartle Bros. Machine Co. Principle of the drying action is shown in the schematic sketch below. Twelve stainless steel blades are arranged in an arc so that the sheet contacts all the blades when stationary. When the sheet is traveling at the speed of 12,000 feet a minute, the adjacent air traveling with the web is shaved into the series of twelve pockets formed by the blades, thereby creating a slight pressure on the blade side of the sheet. This pressure is sufficient to offset the tension in the sheet so that it literally floats over the edges of the blades rather than actually contacting them. Water which adheres to the paper in small beads is brought in contact with the blades and, assisted by the greater affinity it has for stainless than for the wax surface is scooped off and drains into the tank. No marking of the sheet is caused by this method of drying.

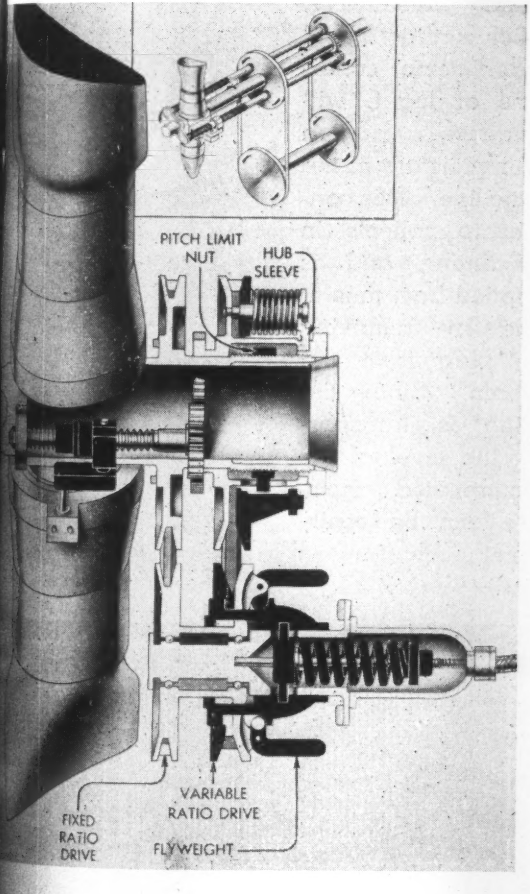
Formerly, the water was removed by passing the sheet over suction bands which had a tendency to mark the sheet. Twenty horsepower on the suction fan was required for the operation which is obviated by the new method. Also, by eliminating the drag on the drive in pulling the sheet over the suction bands the power requirement on the drive is reduced by five horsepower, resulting in a saving of twenty-five horsepower besides producing a better product.





Mechanical control of pitch for constant-speed aircraft propellers utilizes the principle of the variable-ratio V-belt drive. The unit employs





drives for pitch control—one having a fixed and the other being variable through a pair it pulleys as shown in the illustration above, the cowling removed. Pitch change is ef- ed when there is any variation between the s of the two drives.

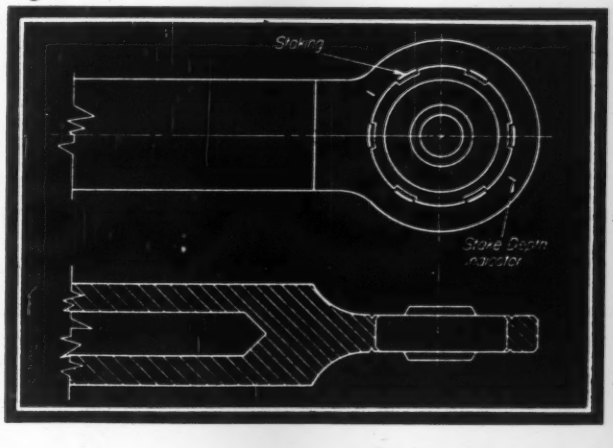
governor shaft is driven by the fixed-ratio e as indicated in the cross-sectional and ematic drawings above. A sleeve that can rotate in either direction relative to the propeller hub is placed around the hub aft the fixed pulley. This sleeve is connected to the pitch mechanism in such a way that, when the sleeve is turned faster than the hub, it increases the pitch of the blades. When turned slower the pitch is reduced. When no relative rotation exists between the two drives there is no pitch change.

Connected to the pitch change sleeve is the variable-ratio drive. Ratio of this drive is controlled by a speed-sensitive governor. When the flyweights are displaced outward the variable-diameter pulley increases the relative speed of the variable unit, causing the sleeve to rotate faster than the hub and consequently increasing the blade pitch. Slower speed of the drive operates the control in reverse and decreases the pitch.

This control, developed by Zimmer-Thomson Corp. has been tested by the Army Air Corps and has been extensively flight tested. According to the designers its life is expected to exceed that of the plane because no deterioration has been detected during tests. Less than ±3 revolutions per minute variation at an engine speed between 2000 and 2500 is the sensitivity of control for this unit.

Staked bearing assemblies, instead of spun, facilitate replacement in service. Staking, as shown below, is accomplished by placing the part containing the bearing between upper and lower dies on a press. Both dies have teeth which force or stake the housing material over the edge of the bearing in six places around its perimeter, thereby anchoring the bearing in its housing. In this method, developed by the Glenn L. Martin Co., both sides are staked simultaneously.

With this system, because only one-third of the distance around the perimeter is staked, it is possible to make replacements twice by staking in a new area each time. Another advantage involves

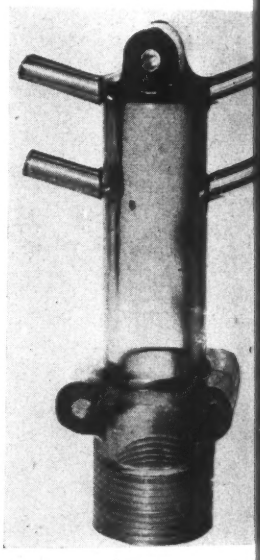


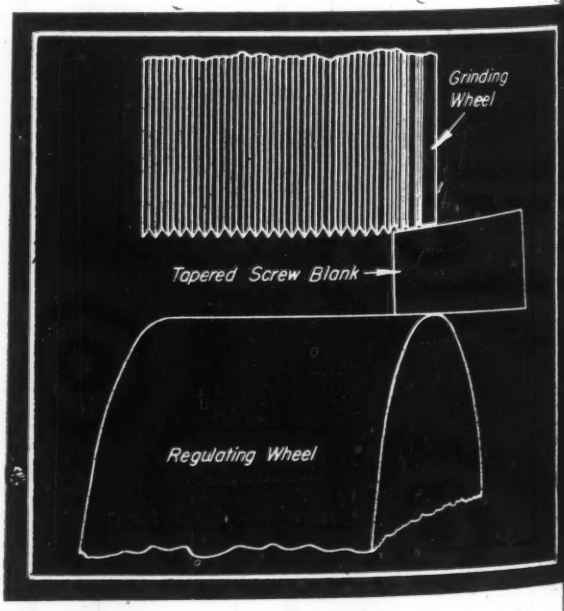
time saving because both sides are staked at one press operation. It has been estimated that this saving will amount to one-third over that required for spinning. Materials which may be staked by this method, as shown above, are the same as those which can be spun.

Centerless grinding of screw threads now meets the requirements of wartime schedules for greater production and increased precision. On a thread grinding machine developed by Landis Tool Company it is possible to grind accurate, smooth threads on hardened-steel setscrew blanks in a single operation. Accuracies of threads within .0005-inch are readily maintained on, for instance, 1/4-inch diameter by 1/2-inch length setscrews produced at the rate of 85 per minute.

Schematic of the principle utilized in the grinder is shown at right. The grinding wheel is selected to have a grain size of sufficient fineness to produce accurate thread profiles and to have a bond that will permit crush-forming of the wheel. Throat or entrance end of the wheel is chamfered with a diamond dressing tool after the thread grooves have been formed. The regulating wheel is a rubber-bonded abrasive wheel with a universal base. A swivel plate permits rotation of this base for adjustment or to grind a taper thread. Relations betwe... regulating and grinding wheels for feeding and s... ing in centerless grinding were discussed in an ... ticle in February, and the design of the Landis m... chine will be treated in a subsequent issue.

Static vent-drain illustrated below is inject... molded from cellulose-acetate-butyrate. Functi... of the fitting, designed by Douglas Aircraft, is tw... fold: It serves as a static pressure equalizer a... as a drain fitting in the pitot static line. Locat... inside the nose of an airplane, the two lower ar... of the part are connected by flexible hose to t... pitot static vents on the outside surface of the fus... lage on each side of the cockpit. The two-upp... arms are connected to the static side of the altime... and air-speed indicator in the pilot's cockpit. ... coming fresh air, which is quite humid under some conditions of flight, will condense a portion of its moisture in the static line. This condensate collects in the fitting and is emptied from time to time by removing the plug from the bottom. Since the fitting is transparent the level of the accumulated moisture can be easily seen.





Improved Methods for Calculating Torsional Vibration

By Robert H. Scanlan
Chief Stress Analyst
Lawrance Aeronautical Corp.

Cut-and-try calculations for free torsional vibration frequencies of complex shaft systems are among the most tedious and time-consuming of the various chores that must be performed in a modern engineering department. Here is presented a method which replaces the repeated approximation procedure with a straightforward computation. Although to one not familiar with matrix techniques the new method may at first glance appear somewhat formidable, the reward of careful study will be the acquisition by the designer of a new and powerful tool

Part 1—Fundamental Mode

METHODS heretofore used for obtaining free torsional vibration frequencies of shafts carrying several concentrated masses such as rotors and gears, or crankshafts with their associated running gear, have been of the "direct-solution" type only for simple systems or where strong simplifying assumptions have been made (1)*. In the latter case the answer often is of little value. The methods of Holzer, Stodola and others, which have been employed to obtain accurate results for the more complex systems, use the technique of repeated approximations.

It is the purpose of this Data Sheet to present a method, based on well-known matrix techniques, which reduces the repeated approximations to a simple, straightforward computation and which does not require unusual simplifying assumptions even in complex cases. The method already has found extensive application in the literature of aircraft wing vibration and flutter (2, 3, 4, 5, 6). Its extension to problems such as are encountered in the design of shaft systems for machines will be discussed here in detail.

To illustrate the contrast of the matrix methods with direct-solution methods, the case of a four-cylinder engine-flywheel system in torsional oscillation will be considered. The cranks, plus reciprocating parts, are replaced, as usual, by "equivalent disks" having moments of inertia calculated from Appendix A. Also, by standard methods (see Appendix B), the cranks are replaced by equivalent plain shafts.

These typical assumptions are common to all analysis methods where varying moments of inertia are not taken into account. (Varying moments of inertia will not be considered here.) A complete discussion of equivalent oscillating systems is given by Wilson (8). The equivalent system to be analyzed appears in *Fig.* 1, where I_0, I_1, I_2, I_3 and I_4 represent equivalent moments of inertia at the disks 0, 1, 2, 3 and 4, L_1, L_2, L_3 and L_4 are the lengths between disks, and C_1, C_2, C_3 and C_4 are the corresponding rigidities or stiffnesses of the connecting shafts between disks.

To proceed further by the direct-solution method would require the setting up of equations of motion for the five disks and the solution of the resulting frequency equation for simple harmonic motion. (That the motion is simple harmonic is assured by the assumption of perfectly elastic shafts.) There being five disks, there will be four actual "modes" of vibration, each with a frequency corresponding to it. This involves solution of a fourth-degree equation (actually a fifth-degree equation with one zero root) for the squared frequencies ω_i^2. While this solution itself is not too difficult, the setting up of such an equation is formidable, and open to computational errors.

It is here that the matrix method to be described carries an essential advantage over the direct solution. It avoids entirely the set-up or solution of higher-degree equations, and is equally applicable to systems of any number of degrees of freedom.

To proceed with the matrix method, the concept of "unit twists" or *torsional influence coefficients* will be introduced. The influence coefficient $\theta(i, j)$ will be defined as the angular displacement of disk *i referred to disk* 0 *as fixed*, when a unit positive torque is applied to disk *j*. It may be noted that the assumption of perfect elasticity makes valid Maxwell's law of reciprocal deflections which, interpreted here, simply means that $\theta(i, j) = \theta(j, i)$.

Since the general formula for angular twist ϕ is

$$\phi = \int_0^L \frac{T dx}{C} \qquad (1)$$

where T is the torque at point x, L is length of shaft,

*References in parentheses are listed at end of article.

and C is torsional rigidity, it follows that if i and j are two adjacent disks distance L apart:

$$\theta(i,j)=\int_0^L \frac{dx}{C}=\frac{L}{C} \quad \ldots\ldots (2)$$

A table of these influence coefficients may be set up for the 5-mass example cited (TABLE I). It may obviously be extended in similar fashion to any number of disks.

TABLE I

Torsional Influence Coefficients for 5-Mass System

(Unit torque applied at j, resultant twist measured at i)

	$j=1$	$j=2$	$j=3$	$j=4$
$i=1$	$\frac{L_1}{C_1}$	$\frac{L_1}{C_1}$	$\frac{L_1}{C_1}$	$\frac{L_1}{C_1}$
$i=2$	$\frac{L_1}{C_1}$	$\frac{L_1}{C_1}+\frac{L_2}{C_2}$	$\frac{L_1}{C_1}+\frac{L_2}{C_2}$	$\frac{L_1}{C_1}+\frac{L_2}{C_2}$
$i=3$	$\frac{L_1}{C_1}$	$\frac{L_1}{C_1}+\frac{L_2}{C_2}$	$\frac{L_1}{C_1}+\frac{L_2}{C_2}+\frac{L_3}{C_3}$	$\frac{L_1}{C_1}+\frac{L_2}{C_2}+\frac{L_3}{C_3}$
$i=4$	$\frac{L_1}{C_1}$	$\frac{L_1}{C_1}+\frac{L_2}{C_2}$	$\frac{L_1}{C_1}+\frac{L_2}{C_2}+\frac{L_3}{C_3}$	$\frac{L_1}{C_1}+\frac{L_2}{C_2}+\frac{L_3}{C_3}+\frac{L_4}{C_4}$

Now, if a torque T_j be applied to disk j the twist of disk i will be $T_j\theta(i,j)$. In general, if torques T_n are applied to each disk the displacement ϕ_i of disk i relative to disk 0 will be:

$$\phi_i-\phi_0=T_1\theta(i,1)+T_2\theta(i,2)+T_3\theta(i,3)+T_4\theta(i,4)\ldots(3)$$

Thus may be written four equations for the system, letting a_{ij} equal $\theta(i,j)$:

$$\left.\begin{aligned}\phi_1-\phi_0&=a_{11}T_1+a_{12}T_2+a_{13}T_3+a_{14}T_4\\ \phi_2-\phi_0&=a_{21}T_1+a_{22}T_2+a_{23}T_3+a_{24}T_4\\ \phi_3-\phi_0&=a_{31}T_1+a_{32}T_2+a_{33}T_3+a_{34}T_4\\ \phi_4-\phi_0&=a_{41}T_1+a_{42}T_2+a_{43}T_3+a_{44}T_4\end{aligned}\right\} \quad \ldots\ldots (4)$$

In the case of free torsional vibration the disk 0, the flywheel, also must rotate. (The entire shaft may be rotating, but this uniform motion can be completely neglected in analyzing the torsional vibration.) Equilibrium of the system when all disks are free to oscillate can be expressed by imposing the obvious condition that the angular momentum (of oscillation) must be zero. This condition is

$$I_0\phi_0+I_1\phi_1+I_2\phi_2+I_3\phi_3+I_4\phi_4=0 \quad \ldots\ldots (5)$$

During free vibration (simple harmonic motion) the torques T_n of Equations 3 and 4 are inertia torques of the various disks:

$$T_n=-I_n\frac{d^2\phi_n}{dt^2}=I_n\omega^2\phi_n \quad \ldots\ldots (6)$$

Equations 4 and 5 then furnish five equations in five unknowns, T_n and the squared frequency ω^2:

$$\phi_1-\phi_0=[\alpha_{11}\phi_1+\alpha_{12}\phi_2+\alpha_{13}\phi_3+\alpha_{14}\phi_4]\omega^2 \quad \ldots\ldots (7a)$$
$$\phi_2-\phi_0=[\alpha_{21}\phi_1+\alpha_{22}\phi_2+\alpha_{23}\phi_3+\alpha_{24}\phi_4]\omega^2 \quad \ldots\ldots (7b)$$
$$\phi_3-\phi_0=[\alpha_{31}\phi_1+\alpha_{32}\phi_2+\alpha_{33}\phi_3+\alpha_{34}\phi_4]\omega^2 \quad \ldots\ldots (7c)$$
$$\phi_4-\phi_0=[\alpha_{41}\phi_1+\alpha_{42}\phi_2+\alpha_{43}\phi_3+\alpha_{44}\phi_4]\omega^2 \quad \ldots\ldots (7d)$$
$$-I_0\phi_0=I_1\phi_1+I_2\phi_2+I_3\phi_3+I_4\phi_4 \quad \ldots\ldots (5)$$

where $\alpha_{ij}=I_j a_{ij}$. By adding I_1 times Equation 7*a*, I_2 times Equation 7*b*, etc., and using Equation 5, ϕ_0 can be expressed in terms of ϕ_1, ϕ_2, ϕ_3 and ϕ_4 and thus can be eliminated. There results a system of equations

$$\left.\begin{aligned}\phi_1&=[b_{11}\phi_1+b_{12}\phi_2+b_{13}\phi_3+b_{14}\phi_4]\omega^2\\ \phi_2&=[b_{21}\phi_1+b_{22}\phi_2+b_{23}\phi_3+b_{24}\phi_4]\omega^2\\ \phi_3&=[b_{31}\phi_1+b_{32}\phi_2+b_{33}\phi_3+b_{34}\phi_4]\omega^2\\ \phi_4&=[b_{41}\phi_1+b_{42}\phi_2+b_{43}\phi_3+b_{44}\phi_4]\omega^2\end{aligned}\right\} \quad \ldots\ldots (8a)$$

where b_{11}, b_{12}, etc., are numerical constants which are functions of the inertia and influence coefficients of the system. Their determination is much simpler when actual numbers are used rather than symbols. This will be demonstrated in the worked example which will follow.

At this point the matrix notation is introduced. In this notation, Equation 8*a* becomes

$$\begin{bmatrix}\phi_1\\ \phi_2\\ \phi_3\\ \phi_4\end{bmatrix}=\begin{bmatrix}b_{11} & b_{12} & b_{13} & b_{14}\\ b_{21} & b_{22} & b_{23} & b_{24}\\ b_{31} & b_{32} & b_{33} & b_{34}\\ b_{41} & b_{42} & b_{43} & b_{44}\end{bmatrix}\times\begin{bmatrix}\phi_1\\ \phi_2\\ \phi_3\\ \phi_4\end{bmatrix}\omega^2 \quad \ldots\ldots (8b)$$

where the "matrix multiplication" indicated on the right between the "square matrix" and the "column matrix" can readily by understood by referring to the conventional form of Equation 8*a*. This type of multiplication is standard for a matrix of n rows and m columns times another of (necessarily) m rows and p columns, the result being a matrix of n rows and p columns. (Above the result will be a one-column matrix).

Since they occur frequently hereafter, examples of matrix multiplication are given here to clarify their meaning. Let M be an arbitrary four-by-four matrix such as

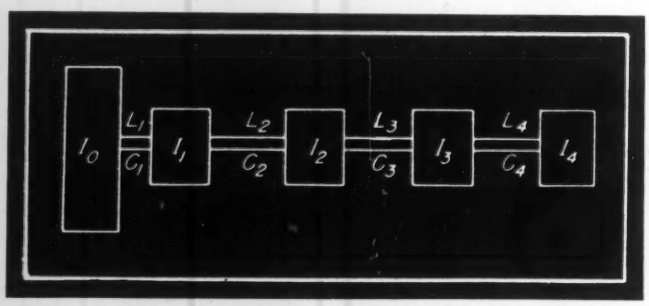


Fig. 1—Typical torsional system for which natural frequencies must be calculated is a four-cylinder engine. Masses are assumed concentrated at center lines of cylinders and flywheel with massless, elastic shafts

$$M=\begin{bmatrix} 6 & 2 & 1 & -3 \\ 5 & 0 & 7 & 5 \\ 2 & 8 & 6 & 2 \\ 3 & 9 & -1 & 4 \end{bmatrix}$$

Postmultiplication of M by an arbitrary column such as

$$C=\begin{bmatrix} 5 \\ 3 \\ 2 \\ 1 \end{bmatrix}$$

is performed as follows:

$$MC=\begin{bmatrix} 6 & 2 & 1 & -3 \\ 5 & 0 & 7 & 5 \\ 2 & 8 & 6 & 2 \\ 3 & 9 & -1 & 4 \end{bmatrix}\begin{bmatrix} 5 \\ 3 \\ 2 \\ 1 \end{bmatrix}=\begin{bmatrix} 35 \\ 44 \\ 48 \\ 44 \end{bmatrix}$$

where $35 = 6 \times 5 + 2 \times 3 + 1 \times 2 - 3 \times 1$

$44 = 5 \times 5 + 0 \times 3 + 7 \times 2 + 5 \times 1$, etc.

Premultiplication of M by an arbitrary row

$$R=[3278]$$

is performed as follows:

$$RM=[3278]\begin{bmatrix} 6 & 2 & 1 & -3 \\ 5 & 0 & 7 & 5 \\ 2 & 8 & 6 & 2 \\ 3 & 9 & -1 & 4 \end{bmatrix}=[66 \quad 134 \quad 51 \quad 47]$$

where $66 = 3 \times 6 + 2 \times 5 + 7 \times 2 + 8 \times 3$

$134 = 3 \times 2 + 2 \times 0 + 7 \times 8 + 8 \times 9$, etc.

Since both ω^2 and the ϕ_n are unknown, solutions will give only relatitve rotations ϕ_n. Thus there will result, up to a constant multiplier, a solution

$$\begin{bmatrix} \overline{\phi}_1 \\ \overline{\phi}_2 \\ \overline{\phi}_3 \\ \overline{\phi}_4 \end{bmatrix}$$

which may conveniently be "normalized" by dividing all $\overline{\phi}$ values by (say) $\overline{\phi}_4$, thus getting 1 as the ϕ_4 entry and values relatitve to this elsewhere. The term *normalize* is a convenience adapted from mathematics to suggest the reduction of all values to a common basis for comparison.

Essence of the *iteration technique* which will be used is to assume such a set of normalized $\overline{\phi}$ values and perform the indicated postmultiplication on the right of Equation 8*b*. Normalizing the result gives a new set of $\overline{\phi}$ values. These are used in the same indicated multiplication, the results normalized, and the process continued until two adjacent cycles give insignificant variation in the results. The process is then said to converge. That this process does converge is proved in standard texts (2, 7). Iteration is an obvious name for this repeating series of operations.

The final value of $\overline{\phi}_4$ used as a divisor in normalizing is a close approximation to $1/\omega^2$ for the fundamental mode. These techniques are illustrated by the following example.

Assume that the four-cylinder engine and flywheel (*Fig.* 1) has the following (simplified) parameters: For flywheel, $I_0 = 600$ pounds-inches-seconds^2, for reciprocating and rotating parts, $I_1 = I_2 = I_3 = I_4 = 100$ pounds-inches-seconds^2, (these need not be equal in general), cylinder spacing $L_2 = L_3 = L_4 = 6$ inches, and distance to flywheel $L_1 = 12$ inches. Shaft stiffnesses are $C_1 = C_2 = C_3 = C_4 = 6 \times 10^7$ pounds-inches^2.

Influence coefficient table (TABLE I) becomes, on substituting these values:

Values of $\theta(i, j) \times 10^7$

	$j=1$	$j=2$	$j=3$	$j=4$
$i=1$	2	2	2	2
$i=2$	2	3	3	3
$i=3$	2	3	4	4
$i=4$	2	3	4	5

In general, of course, these values will not be round numbers. Equations 7 are then

$$\left.\begin{aligned} \phi_1-\phi_0&=[2\phi_1+2\phi_2+2\phi_3+2\phi_4]\omega^2\times10^{-5} \\ \phi_2-\phi_0&=[2\phi_1+3\phi_2+3\phi_3+3\phi_4]\omega^2\times10^{-5} \\ \phi_3-\phi_0&=[2\phi_1+3\phi_2+4\phi_3+4\phi_4]\omega^2\times10^{-5} \\ \phi_4-\phi_0&=[2\phi_1+3\phi_2+4\phi_3+5\phi_4]\omega^2\times10^{-5} \end{aligned}\right\} \quad \ldots\ldots\ldots\ldots [a]$$

and Equation 5 becomes

$$-600\phi_0=100(\phi_1+\phi_2+\phi_3+\phi_4) \quad \ldots\ldots\ldots\ldots\ldots\ldots [b]$$

Following the same step as from Equation 7 to Equation 8*a*, there results the following group of equations:

$$\begin{aligned} 100(\phi_1-\phi_0)&=[2\phi_1+2\phi_2+2\phi_3+2\phi_4]\omega^2\times10^{-3} \\ 100(\phi_2-\phi_0)&=[2\phi_1+3\phi_2+3\phi_3+3\phi_4]\omega^2\times10^{-3} \\ 100(\phi_3-\phi_0)&=[2\phi_1+3\phi_2+4\phi_3+4\phi_4]\omega^2\times10^{-3} \\ 100(\phi_4-\phi_0)&=[2\phi_1+3\phi_2+4\phi_3+5\phi_4]\omega^2\times10^{-3} \end{aligned}$$

Making use in this of Equation *b* gives

$$100(\phi_1+\phi_2+\phi_3+\phi_4)-400\phi_0=[8\phi_1+11\phi_2+13\phi_3+14\phi_4]\omega^2\times10^{-3}$$

Making use in this of Equation *b* gives

$$-600\phi_0-400\phi_0=[8\phi_1+11\phi_2+13\phi_3+14\phi_4]\omega^2\times10^{-3}$$

or

$$\phi_0=-[.8\phi_1+1.1\phi_2+1.3\phi_3+1.4\phi_4]\omega^2\times10^{-5}$$

Substituting this value of ϕ_0 in Equations *a*,

$$\phi_1=[1.2\phi_1+.9\phi_2+.7\phi_3+.6\phi_4]\omega^2\times10^{-5}$$
$$\phi_2=[1.2\phi_1+1.9\phi_2+1.7\phi_3+1.6\phi_4]\omega^2\times10^{-5}$$
$$\phi_3=[1.2\phi_1+1.9\phi_2+2.7\phi_3+2.6\phi_4]\omega^2\times10^{-5}$$
$$\phi_4=[1.2\phi_1+1.9\phi_2+2.7\phi_3+3.6\phi_4]\omega^2\times10^{-5}$$

thus eliminating ϕ_0. Evaluation of Equation 8*b* is:

$$\begin{bmatrix}\phi_1\\ \phi_2\\ \phi_3\\ \phi_4\end{bmatrix}=\begin{bmatrix}1.2 & .9 & .7 & .6\\ 1.2 & 1.9 & 1.7 & 1.6\\ 1.2 & 1.9 & 2.7 & 2.6\\ 1.2 & 1.9 & 2.7 & 3.6\end{bmatrix}\begin{bmatrix}\phi_1\\ \phi_2\\ \phi_3\\ \phi_4\end{bmatrix}\omega^2\times10^{-5}$$

The iteration process by postmultiplication on this is given in TABLE II. Matrix multiplication to get the second column in TABLE II from the first is as follows:

$$\begin{bmatrix}1.2 & .9 & .7 & .6\\ 1.2 & 1.9 & 1.7 & 1.6\\ 1.2 & 1.9 & 2.7 & 2.6\\ 1.2 & 1.9 & 2.7 & 3.6\end{bmatrix}\begin{bmatrix}.4\\ .6\\ .8\\ 1.0\end{bmatrix}=\begin{bmatrix}2.60\\ 4.58\\ 6.38\\ 7.38\end{bmatrix}$$

The process involved in going from the second to the third column is as follows, correct to four decimal places: 2.60/7.38 = .3523; 4.58/7.38 = .6206; 6.38/7.38 = .8645; 7.38/7.38 = 1.

The entire process is repeated, starting with the third column for postmultiplication, instead of the first, to get successive columns. Each odd-numbered column is a closer approximation to the actual values of ϕ which define the mode. To get ϕ_0, the results in the final column of TABLE II are used in Equation 5, giving $\phi_0 = -.4604$.

Frequency equation is $10^5/\omega_1^2=7.4441$, from which $\omega_1 = 115.9$ radians per second, and the frequency is $f_1 = (60/2\pi)\omega_1 = 1106.8$ cycles per minute. The iteration could have been carried out by premultiplication of the square matrix of Equation 8*b* by a row matrix also, but this does not yield the mode; only the frequency is given in this case.

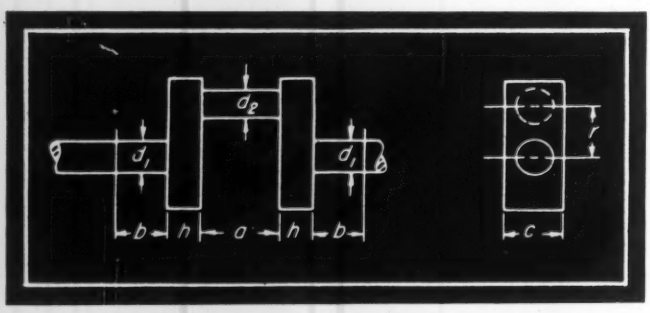


Fig. 2—Basic dimensions from which equivalent length and stiffness of a crank throw are calculated

By the nature of the iteration process as set up in the form of TABLE II there is assurance that this is the fundamental frequency (7), and the final column gives the relative ϕ values for the fundamental mode.

Methods for obtaining higher modes and frequencies will be covered in Part II of this Data Sheet, to be published next month.

Appendix A

Formula for equivalent moment of inertia of rotating and reciprocating masses is as follows (Ref. 1, Page 273):

$$I=\left\{M_1+\tfrac{1}{2}M\left(1+\frac{r^2}{4l^2}\right)\right\}r^2$$

where M_1 = Mass assumed concentrated on crankpin to replace mass of rotating parts
M = Mass of reciprocating parts (piston, etc.)
r = Crank radius
l = Connecting rod length.

Appendix B

Formula for equivalent length L and stiffness C of crankshaft, *Fig.* 2, is as follows (Ref. 1, Page 270):

$$L=C\left(\frac{2b_1}{C_1}+\frac{a_1}{C_2}+\frac{2r}{B}\right)$$

where $b_1 = b + .45h$ = "Effective length" of journal
$a_1 = a + .9h$ = "Effective length" of crankpin
$C_1 = (\pi d_1^4 G)/32$ = Torsional rigidity of journal
$C_2 = (\pi d_2^4 G)/32$ = Torsional rigidity of crank
$B = (h\,c^3\,E)/12$ = Flexural rigidity of web
$G = 11.5 \times 10^6$ pounds per square inch = Shear modulus of steel
$E = 30 \times 10^6$ pounds per square inch = Modulus of elasticity of steel.

REFERENCES

1. Timoshenko, S.—*Vibration Problems in Engineering* (2nd Ed.), Van Nostrand, New York, 1937.
2. Frazer, Duncan and Collar—*Elementary Matrices*, Cambridge, 1938.
3. Biot, M. A.—"Vibration Analysis of a Wing Carrying Large Concentrated Weights", GALCIT Report No. 1, 1940.
4. Lieber, Paul—"An Iteration Method for Calculating the Latent Roots of a Flutter Matrix", Report S.M. 3855, Douglas Aircraft Co., Santa Monica, California, Sept., 1942.
5. Wasserman, Lee—Comments on paper by E. F. Critchlow, *S.A.E. Journal*, Aug., 1944.
6. Rosenbaum, R., and Scanlan, R. H.—"Influence Coefficients in Stress and Vibration Analysis", Paper presented at the Summer Annual Meeting, I.A.S., Los Angeles, July, 1944.
7. v. Karman, Th., and Biot, M. A.—*Mathematical Methods in Engineering*, McGraw-Hill, New York, 1940.
8. Wilson, W. K.—*Practical Solution of Torsional Vibration Problems* (2nd Ed.), Wiley, New York, 1940.

TABLE II

Calculation of Relative Vibration Amplitudes in First Mode

Assumed mode	Col. 2	Col. 3	Col. 4	Col. 5	Col. 6	Col. 7	Col. 8	Col. 9	Col. 10	Col. 11	Col. 12	Col. 13	Mode ϕ
.4	2.60	.3523	2.186	.2901	2.113	.2829	2.100	.2819	2.09793	.281804	2.097683	.281792	.2818
.6	4.58	.6206	4.672	.6200	4.601	.6161	4.582	.6151	4.57883	.615051	4.578408	.615040	.6150
.8	6.38	.8645	6.536	.8673	6.468	.8661	6.449	.8658	6.44463	.865674	6.444082	.865665	.8657
1.0	7.68	1.0000	7.536	1.0000	7.468	1.0000	7.449	1.0000	7.44463	1.000000	7.444082	1.000000	1.0000

MACHINE DESIGN'S

MATERIALS WORK SHEET

FILING NUMBER
31.00

Wool Felts

Standard SAE Grades

AVAILABLE IN: (Grades F-1, F-2, F-3, F-50, F-51, and F-55) Long rolls or cuts 60 inches wide. All other grades in long rolls or cuts 72 inches wide.
All grades can be procured in a wide variety of ready-to-use parts such as oil, grease and dust seals, washers, gaskets, round wicking, etc.

ANALYSES

SAE No.	Trade Designation	Color	Fiber Basis[1] Wool Fiber (%)	Fiber Basis[1] Cotton Fiber (%)	Chemical Requirements: Wool Content (min %)	Carbon Tetrachloride Soluble (max %)	Water Soluble (max %)	Ash (max %)	Nonfibrous Impurities (max %)
F-1	Back Check	White	100	...	95	2.5	2.5	1.0	3.0
F-2	Back Check	Any Color[2]	100	...	95	2.5	2.5	1.5	4.0
F-3	Back Check	Gray	95-100	5	90	2.5	3.0	1.5	4.5
F-5	Extra Firm Pad	White	100	...	95	2.5	2.5	1.0	3.0
F-6	Extra Firm Pad	Gray	100	...	92	2.5	2.5	1.5	4.5
F-7	Extra Firm Pad	Gray	85-95	5-15	80	3.0	4.0	2.0	7.0
F-10	Firm Pad	White	100	...	95	2.5	2.5	1.0	3.0
F-11	Firm Pad	Gray	100	...	92	3.0	2.5	2.0	4.5
F-12	Firm Pad	Gray	90-95	5-10	85	4.0	2.5	2.0	6.5
F-13	Firm Pad	Gray	80-85	15-20	75	4.0	4.0	2.5	8.0
F-15	Firm Pad	Gray	60-65	35-40	55	4.0	5.0	4.0	9.0
F-26	Soft Pad	Gray	50-55	45-50	45	8.0	6.0	5.0	14.0
F-50	Ball Bearing Felt	White	100	...	95	2.5	2.5	1.0	3.0
F-51	Ball Bearing Felt	Gray	100	...	92	2.5	2.5	1.5	4.5
F-55	Lining	Gray or Black	80-85	15-20	75	4.0	4.0	3.0	8.0

[1] Nominal fiber content indicates the percentage of wool and cotton fiber used in fabricating the several grades of felt and includes traces of other fibers and impurities present in the raw material. Inclusions of traces of cotton and other fibers, residual wool fats and oils or soaps used in processing may reduce the actual wool fiber content on analysis to values shown under "Chemical Requirements."

[2] Except gray and black.

PROPERTIES

BREAKING STRENGTH‡
(min., psi)

F-1	500
F-2	500
F-3	400
F-5	400
F-6	275
F-7	250
F-10	225
F-11	200
F-12	100
F-13	75
F-15	75
F-50	500
F-51	300
F-55	200

SPLITTING RESISTANCE§
(lb per 2-in. width, min.)

F-1	33
F-2	30
F-3	24
F-5	18
F-6	16
F-7	12
F-10	8
F-11	6
F-12	3
F-13 and F-15	2

§S.A.E. values. Splitting resistance is not applicable to felts where the thickness is less than $\frac{3}{16}$-inch. For materials less than $\frac{1}{16}$-inch thick, breaking strength only is recommended as an indicative test.

AVERAGE ELONGATION†
(per cent, within nondestructive limits)

Tensile Load (psi)	SAE Grades F-1 (20.7)*	F-3 (20.1)*	F-5 (15.2)*	F-10 (11.2)*	F-11 (12.0)*	F-15 (11.0)*
25	5	6	6	14	12	14
50	8	10	10	24	19	24
100	13	16	16	33	37	35
200	20	27	27	51	53	..
300	28	39	37	..	..	..
400	35	51	47	..	..	..
500	44	..	..	..	..	..
600	51	..	..	..	..	..

*Density of test piece expressed in pounds per cubic foot.

†From the paper "Mechanical Properties and Uses of Wool Felt", presented at the recent A.S.M.E. annual meeting in New York.

‡Values conform to S.A.E. standards.

MACHINE DESIGN is pleased to acknowledge the collaboration of The Felt Association Inc., and the Society of Automotive Engineers Inc. in this presentation.

Note: This Work Sheet covers only those felts standardized by the S.A.E. Other types, compounded and processed to meet specific engineering requirements, also are available.

OIL ABSORPTION*

(per cent increase in weight)

F-1	215
F-2	230
F-3	190
F-6	370
F-7	290
F-10	450
F-11	475
F-12	500
F-13	525
F-15	500
F-26	625
F-50	225

*Percentage increase in weight when impregnated in oil at 76 F. Values are average for SAE oils 10 to 50.
Data supplied by American Felt Company.

OIL WICKING RATES†

(inches in stated elapsed time)

SAE Grade	Elapsed Time (minutes) 5	10	15	30	60	90	120
F-1	.5	.7	.8	1.1	1.6	2.0	2.2
F-3	.5	.7	1.0	1.3	1.7	1.8	2.1
F-5	.6	.8	1.0	1.3	1.7	1.9	2.0
F-11	.6	.8	1.1	1.7	2.0	2.2	2.4
F-15	.6	.9	1.1	1.4	1.6	1.8	1.9

Note: Oil used for wicking tests was S.A.E. No. 20 at 70° F.

COEFFICIENT OF FRICTION

Surfaces	Coefficient of Sliding Friction
Felt against smooth wood	.32 to .42
Felt against glass	.22 to .29
Felt against smooth metal	.20 to .24
Felt against oiled metal	.18 (for No. 80 oil)

Note: Measurements of the coefficient of friction between various densities of felt show that against the same surface the coefficient varies only within narrow limits. If oil is used as a lubricant between the surfaces the coefficient is affected more by the viscosity of the oil than by the density of the felt.
Data supplied by The Felters Company.

DEFORMATION IN COMPRESSION†

(approx., inches per inch)

Load (psi)	SAE Grades F-1	F-3	F-5	F-10	F-11	F-15
10	.04	.10	.15	.30	.25	.27
20	.08	.12	.20	.38	.31	.36
30	.11	.16	.25	.42	.37	.40
40	.15	.19	.28	.46	.41	.43
50	.19	.22	.31	.48	.44	.46
60	.21	.24	.34	.50	.46	.49
70	.24	.26	.36	.52	.48	.50
80	.26	.28	.38	.53	.51	.52
90	.28	.30	.40	.55	.52	.53
100	.29	.31	.41	.56	.53	.55
120	.31	.34	.43	.58	.56	.57
140	.33	.35	.45	.59	.57	.58
160	.35	.37	.47	.61	.59	.60
180	.36	.38	.48	.62	.60	.61
200	.37	.40	.49	.63	.61	.62
220	.39	.41	.51	.635	.62	.625
240	.40	.42	.52	.64	.63	.635
260	.41	.43	.53	.65	.63	.64
280	.42	.44	.54	.66	.64	.65
300	.43	.45	.55	.67	.65	.66
340	.45	.47	.56	.68	.66	.67
380	.46	.48	.57	.70	.68	.69
420	.48	.50	.58	...	.70	...
460	.49	.51	.59	...	...	...
500	.51	.52	.61	...	...	...

Note: Above figures are only indicative. Not to be used for definite specifications in design.

WATER SHRINKAGE—THICKNESS SWELL†

(by ASTM D461 test method)

SAE Grade	Per Cent Shrinkage Length	Width	Per Cent Swell in Thickness
F-1	—0.2	—2.5	15.1
F-3	—1.5	—1.1	17.2
F-5	—0.2	—3.0	14.7
F-10	—0.6	—1.8	13.7
F-11	—1.4	—3.5	14.9
F-15	—2.0	—1.1	17.2

Note: These figures are the average record of tests made on forty-two representative standard S.A.E. felt qualities and densities ¼ and ½-inch thick.

THERMAL CONDUCTIVITY†

(for 1-inch felt)

SAE Grade	Btu/hr/sq ft/deg F
F-1	.36
F-3	.35
F-5	.30
F-11	.23
F-15	.24

VIBRATION DAMPING CHARACTERISTICS

Natural Frequencies at Loading Limits

(in cycles per second for felts 1-in. thk.)

Load (psi)	SAE Grades F-2 (21.3)*	F-6 (16.3)*	F-11 (11.4)*
2	..	..	30
4	..	..	26
6	..	..	24
8	..	29	..
10	..	27	..
15	..	25	..
20	31	..	..
25	29	..	..
30	27	..	..
35	26	..	..

*Density, expressed in pounds per cubic foot.
Note: Loading beyond the above limits will result in excessive deformation, increased stiffness of the system, and decreased vibration isolation.

ABRASIVE WEAR†

(per cent decrease in thickness)

SAE Grade	Applied Loads (oz per sq in.) 3.8	7.0	10.2
F-1	1.0	2.3	2.1
F-3	2.1	2.1	3.0
F-5	3.0	3.0	6.0
F-15	4.1	7.6	18.5

Note: Data apply to rubbing felt on felt, 10,000 eight-inch strokes.

HARDNESS-DENSITY RELATIONSHIP

(approx., S.A.E. standard felts)

Density (lbs per cu ft)	Shore "A" Durometer Hardness
9	5
11.5	10
13.0	15
14.0	20
16.5	25
17.5	30
19.5	35
22.0	40
24.5	45

Note: These data are based on one series of tests. They are not to be considered standard or suitable for definite specification.

COMPRESSION SET AND DEFORMATION†

(average)

Deformation (per cent)	SAE Grades F-1 (20.7)* Load (psi)	F-1 Comp. Set (per cent)	F-3 (20.1)* Load (psi)	F-3 Comp. Set (per cent)	F-5 (15.2)* Load (psi)	F-5 Comp. Set (per cent)	F-10 (11.2)* Load (psi)	F-10 Comp. Set (per cent)	F-11 (12.0)* Load (psi)	F-11 Comp. Set (per cent)	F-15 (11.0)* Load (psi)	F-15 Comp. Set (per cent)
10	21	1	13	1	6	1	4	1	4	1	3	1
20	45	2	35	3	17	2	7	2	7	2	6	2
30	108	4	93	6	41	4	9	3	15	4	10	4
40	246	7	210	9	91	6	24	6	34	6	26	6
50	480	11	421	12	213	10	58	9	72	9	64	10
60	1,297	23	1,223	24	458	17	151	13	173	13	162	15
80	27,890	53	21,180	55	7,746	53	2,382	40	2,625	42	3,416	42

Note: Compression set is expressed as the percentage of the difference between original and final thickness to the original thickness.
*Density, expressed in pounds per cubic foot.

CHARACTERISTICS

Made without spinning, weaving or knitting, felt is a matted fabric formed from wool and other fibers by a suitable combination of mechanical work, pressure, heat and moisture. It does not ravel or fray and can be cut to many different shapes for gaskets, washers, wicks, etc. Available in widely varying degrees of hardness, some of its grades are hard enough to turn on a lathe; others are soft enough to sew. It offers exceptional resistance to deterioration, being little affected by age and exposure to weather. Being homogeneous in structure, its physical properties remain unchanged as it wears down in thickness. It is highly oil-absorbent and is not injured by long, continued saturation. It possesses good resilience, consequently can be compressed and released thousands of times without permanent deformation. It has excellent qualities of insulation against cold or heat, sound and vibration. Obtainable in almost any color desired, it can be made water-repellent, water-absorbent or flame-resistant by appropriate chemical treatments.

APPLICATIONS

SAE F-1: Is suitable for oil retention in installations where the felt is not compressed; for feeding low-viscosity or light oil, and where unusual strength and hardness are required. Ink rollers, bushings, lubricating wicks, axle oil-retaining washers, polishing wheels, door bumpers, embossing plates, machine pads, washers, and parts where wear and resistance to abrasion are required.

SAE F-2 AND F-3: Recommended for same general purposes as grade F-1 and where felt of slightly lower quality is satisfactory. Some typical applications are: Packing, oil and water gaskets, chassis antisqueak members, tank padding, felt feet, vibration damping members, and washers.

SAE F-5, F-6 AND F-7: Recommended for dust shields, wipers, grease-retaining washers, wicks, oil filters, strips, gaskets, and in uses where a resilient felt is required.

SAE F-10, F-11 AND F-12: Are recommended for grease and oil retention where the felt is confined and compressed in assembly. Ball bearing seals, free-flow lubricating wicks, gaskets, axle washers, moistening felts, grease seals, film polishing pads, sanding machine drums, and packing. Also recommended for dust shields in applications where, because of less severe operating conditions, F-5, F-6 or F-7 are not required.

SAE F-13 AND F-15: For mechanical purposes where abrasion and wear are not important factors. Recommended for sound deadening, chassis strips, spacers, dust shields, pedal pads, radiator packing, antisqueak strips, insulating, packing, and stamp pads.

SAE F-26: Is suitable for packing, padding and insulation when held in place between other materials. Should not be used for mechanical purposes.

SAE F-50: For mechanical purposes where an accurate, thin, smooth, high-grade felt is required. Recommended for ball and roller bearing oil-retainer washers and small dust-excluding washers.

SAE F-51: Same general uses as F-50 but in applications where tolerances and length of life are not as important. Also for thin cut parts such as gaskets and liners.

SAE F-55: Antisqueak strips. Also for lining when cemented to fiber board or metal panels.

FABRICATION

Felt can be die-cut, punched, skived, chiseled, turned, scarfed and ground. In general, these operations are similar to those commonly performed on leather, sheet rubber, fabricated gaskets of various compositions, and other nonmetallic materials. Sheet felts are cut with woodworking tools. Intricate parts often can be built up of separate pieces, assembled by sewing or with adhesives.

RESISTANCE PROPERTIES

Wool felts are not deteriorated by oils, greases, solvents, moderate heat, extreme cold, dryness, sunlight, and ozone. When soaked in distilled, tap, chlorinated, or sea water at room temperature, wool felts will swell moderately in thickness and shrink slightly in length and width. Also breaking strength will be lowered by about 25 per cent. However, when redried after such soaking, they recover their initial strength. Their dimensional stability and physical properties are not affected appreciably by extended exposure to lubricating oils and organic solvents. In concentrations up to 14 degrees Baume, strong inorganic acids such as sulphuric and hydrochloric are noninjurious. It has been observed, however, that nitric acid in 14 degrees Baume concentration decreases the breaking strength of wool felt by 30 to 40 per cent after a 5-day immersion. Strong bases and alkalies are naturally injurious.

In regard to temperature, when wool felts are used above 212 degrees Fahr. the wool loses its hygroscopic moisture, and breaking strength, elongation and splitting resistance are adversely affected. Wool felt does not support combustion and can be flame-proofed to render it fire resistant. For most applications it is regarded as a permanent material, stable under normal conditions of use.

DATA ON STOCK FORMS

Permissible Tolerances in Thickness‡ (inches)

SAE	Nominal Thickness (inches)											
	1/8		3/16		1/4		5/16		3/8		1/2	
No.	min	max	min	max	min	max	min	max	min	max	min	max
F-1	.112	.137	.173	.201	.235	.265	.296	.329	.358	.392	.480	.520
F-2	.112	.137	.173	.201	.235	.265	.296	.329	.358	.392	.480	.520
F-3	.112	.137	.173	.201	.235	.265	.296	.329	.358	.392	.480	.520
F-5	.112	.137	.172	.203	.231	.268	.291	.334	.350	.399	.470	.530
F-6	.112	.137	.172	.203	.231	.268	.291	.334	.350	.399	.470	.530
F-7	.112	.137	.172	.203	.231	.268	.291	.334	.350	.399	.470	.530
F-10	.105	.145	.164	.211	.223	.277	.282	.342	.341	.408	.460	.540
F-11	.105	.145	.164	.211	.223	.277	.282	.342	.341	.408	.460	.540
F-12	.100	.150	.158	.217	.217	.283	.275	.350	.333	.416	.450	.550
F-13	.100	.150	.158	.217	.217	.283	.275	.350	.333	.416	.450	.550
F-15	.100	.150	.158	.217	.217	.283	.275	.350	.333	.416	.450	.550
F-26	.085	.165	.142	.233	.198	.302	.255	.370	.311	.438	.425	.575

Weight Requirements‡ (lbs per sq yd)

SAE	Nominal Thickness (inches)											
	1/8		3/16		1/4		5/16		3/8		1/2	
No.	min	max	min	max	min	max	min	max	min	max	min	max
F-1	1.9	2.1	2.85	3.15	3.8	4.2	4.75	5.25	5.7	6.3	7.6	8.4
F-2	1.9	2.1	2.85	3.15	3.8	4.2	4.75	5.25	5.7	6.3	7.6	8.4
F-3	1.8	2.1	2.71	3.15	3.61	4.19	4.50	5.24	5.41	6.29	7.21	8.39
F-5	1.45	1.61	2.18	2.40	2.91	3.21	3.63	4.01	4.36	4.82	5.81	6.43
F-6	1.45	1.61	2.18	2.40	2.91	3.21	3.63	4.01	4.36	4.82	5.81	6.43
F-7	1.42	1.64	2.12	2.46	2.83	3.29	3.53	4.11	4.25	4.93	5.66	6.58
F-10	1.01	1.11	1.51	1.67	2.02	2.24	2.53	2.79	3.03	3.35	4.04	4.46
F-11	0.98	1.14	1.47	1.71	1.97	2.29	2.48	2.84	2.95	3.43	3.93	4.57
F-12	0.98	1.14	1.47	1.71	1.97	2.29	2.48	2.84	2.95	3.43	3.93	4.57
F-13	0.98	1.14	1.47	1.71	1.97	2.29	2.48	2.84	2.95	3.43	3.93	4.57
F-15	0.98	1.14	1.47	1.71	1.97	2.29	2.48	2.84	2.95	3.43	3.93	4.57
F-26	0.86	1.04	1.28	1.56	1.71	2.09	2.07	2.53	2.56	3.14	3.42	4.18

Weight Requirements—Extra Thin Grades‡ (oz per yd, 60 inches wide)

SAE	Nominal Thickness (inches)							
No.	.050	.055	.060	.065	.070	.075	.080	.085
F-50	20	22	24	26	28	30	32	34
F-51	20	22	24	26	28	30	32	34
F-55	15.5	17	18.5	20	21.5	23	24.5	26

MATERIAL DESIGNATIONS

S.A.E.	U.S. Army	U.S. Navy	Felt Assoc.
F-1	8-15G #21	27F7d Type F-1	B-30
F-2	.....	.....	B-35
F-3	8-15G #13, 14, 15, 16	27F7d Type F-3	B-40
F-5	.....	27F7d Type F-5	C-30, C-35
F-6	.....	.....	C-40, C-45
F-7	8-15G #7, 23, 24	27F7d Type F-7	C-50, C-55
F-10	8-15G #28	.....	D-10
F-11	.....	.....	D-20
F-12	.....	.....	D-30
F-13	8-15G #3, 12	27F7d Type F-13	D-40
F-15	8-15G #5, 6, 22	.....	D-50
F-26	8-15G #27	27F7d Type F-26	D-56
F-50	.....	.....	H-10
F-51	.....	.....	G-10
F-55	.....	.....	G-30

PROFESSIONAL VIEWPOINTS

MACHINE DESIGN welcomes comments from readers on subjects of interest to designers. Payment will be made for letters and comments published

"... developments not inventions"

To the Editor:

G. V. Woodling's article, "Group Invention Must Be Upheld" published in the February issue of MACHINE DESIGN, affords a better understanding of why our patent system is contradictory and discouraging to inventive activity. It is a clear picture of a situation existing in the United States, whereby a group of scientifically and technically illiterate judges try to invalidate and discredit the universally acknowledged and historically approved invention of Mr. Marconi.

The Supreme Court judge's opinion presupposes that there is invention in group activity, with or without the inventor. Invention and development are mixed and treated as the same thing in order to deny a patent to the individual and original inventor. In spite, however, of the absurd misunderstanding of the nature of the birth of an invention, the Supreme Court properly refuses to grant a patent on so-called "group development". Inventing something is entirely different from developing something. Successful development by a group requires either the cooperation of the inventor, or the services of competent scientists who can reduce the invention to practice. However, it is never the group which makes the invention but someone inside or outside the group.

It is evident that the main thought behind the ears of a judge who writes such an opinion is to grant a patent neither to the individual inventor nor to the corporation employing the group. This is the theory of the collectivist countries where both the individual and the corporation are subordinate to the state.

—DR. ELEK K. BENEDEK
Chicago, Ill.

"... journal finish is critical"

To the Editor:

I would like to compliment you on the excellent article "Surface Finish—Key to Bearing Life", by E. L. Hemingway, published in the November issue of MACHINE DESIGN. This is certainly a good round-up of the subject and one of the most thought-provoking articles I have read in some time. Particularly interesting was the comment made by the author which, if I interpret it properly, was that scratches on a journal parallel or substantially parallel to the axis of the journal are far less harmful than annular scratches of the same depth. This is so contrary to the prevailing ideas on the subject that I believe some further elaboration would be in order.

—R. E. ORTON, *Chief Engineer*
Acme Steel Co.

To the Editor:

Mr. Orton's comments on our ideas concerning the effects of scratch direction do not surprise me. I feel certain that in many cases we have been imitating the ostrich in using a "codirectional" finish. We have been hiding the damage done in operation by thus placing the original scratches in the exact direction of subsequent scoring. This is particularly true on flat reciprocating bearings where the surfaces are not efficiently separated by oil-film pressure.

Judgment must be used, of course, in each individual case because of the employment of a compromise between the possibility of pure cutting by roughness and the necessity for breaking up minute welding actions. For instance, I would very definitely use a codirectional finish where a piston rod operates through a nonmetallic packing because a cross-directional scratch would act as a dull tool to wear away the packing. In addition, there would be no possibility of welding action taking place. In all probability an extremely smooth finish would be still better for such piston rods.

Another point I would like to make is that the exact angle of the scratch does not seem to be very important. A deviation of only a few degrees from the direction of motion effectively breaks the continuity of any welding action and distributes the oil across the ridge tips.

Lest too much importance be attached to scratch pattern, I would also like to point out the fact that roughness or depth of pattern is much more harmful than variation in direction of scratch. In the article it was shown that a decrease in scratch depth from 10 to 2 microinches showed an increase in load-carrying capacity of 66 per cent. Reduction of waviness and production of very accurate dimensional shape also are much more important considerations. The difference in load-carrying capacity of commercially finished shafts and bushings and those produced with an accuracy measured in millionths of an inch is amazing.

—E. L. HEMINGWAY, *Chief Metallurgist*
International Detrola Corp.

ASSETS to a BOOKCASE

Vibration Analysis

By N. O. Myklestad, research associate, in charge of vibration and flutter, Guggenheim Aeronautical laboratory, California Institute of Technology; published by McGraw-Hill Book Co. Inc., New York; 300 pages, 5¼ by 8¼ inches, clothbound; available through MACHINE DESIGN, $3.50 *postpaid.*

Shaking or exciting forces which upset the equilibrium of a machine or machine part produce the phenomenon well known to engineers—vibration. Such forces and motions are always present in moving machines, and in most cases they are undesirable. Detrimental effects that they can produce include increased stresses of machine parts; interference with proper functioning of the machine itself and other machines and instruments in the vicinity; physiological discomfort, particularly when the vibration is a noisy one; and loss of mechanical energy due to damping forces, which are always present.

This book on vibration analysis has been written mainly to provide the mechanical engineer with the best methods of solving the most important practical vibration problems occurring in modern machine designs. Special problems of considerable complexity are treated in the text primarily for their value to vibration specialists.

Introductory chapters of the book place principal emphasis on helping the reader to understand clearly the various phases in the occurrence of vibration. Of particular interest are the simple tabular methods, which have been developed by the author, for solving most of the important vibration problems that arise in modern engineering involving more than one degree of freedom. All sections are well illustrated with examples and drawings. A variety of questions and answers are included in order to emphasize the most important points covered.

□ □ □

Air Compressors

By Eugene W. Feller, operating engineer, member of National Association of Power Engineers; published by McGraw-Hill Book Co. Inc., New York; 460 pages, 5¼ by 8¼ inches, clothbound; available through MACHINE DESIGN, $4.50 *postpaid.*

Sound engineering in the field of compressed air and compressors is primarily evolved from a background of knowledge which tempers the theoretical with the practical. Although many books are available on the principles and thermodynamics involved in air compression, the material is highly theoretical and when viewed from the practical side of the picture offers little or no help. However, the inherent advantages of air power are creating an expansion of industrial applications and practical knowledge on economical and efficient air compression is at a premium.

The author covers reciprocating, rotary, centrifugal and axial-flow compressors of all makes in his text. Various designs of each type are thoroughly discussed, giving a mechanical description of the component parts and an operating explanation, with curves and drawings showing the working principle of each of the mechanisms. Valve design, piston rings, packings, bearings, vanes, lubrication, oil selection, etc., are representative of the many points covered. Although each is necessarily incident to compressors or compressing equipment, any machine designer may find embodied in these tried and proved devices basic engineering principles which might well aid in the development or improvement of equipment used in some seemingly dissociated field.

□ □ □

Guide to Weldability of Steels

Published by the American Welding Society, New York; 89 pages, 6 by 9 inches, paperbound; available through MACHINE DESIGN, $1.00 *postpaid.*

High uniformity of materials used in the fabrication of welded machinery parts and the safety of such parts under extreme operating conditions have tended to widen the field of weldments into what is now perhaps one of the most significant stages of growth since their inception. Further improvements no doubt are still to come as the science of controlled welding is made a practical everyday accomplishment.

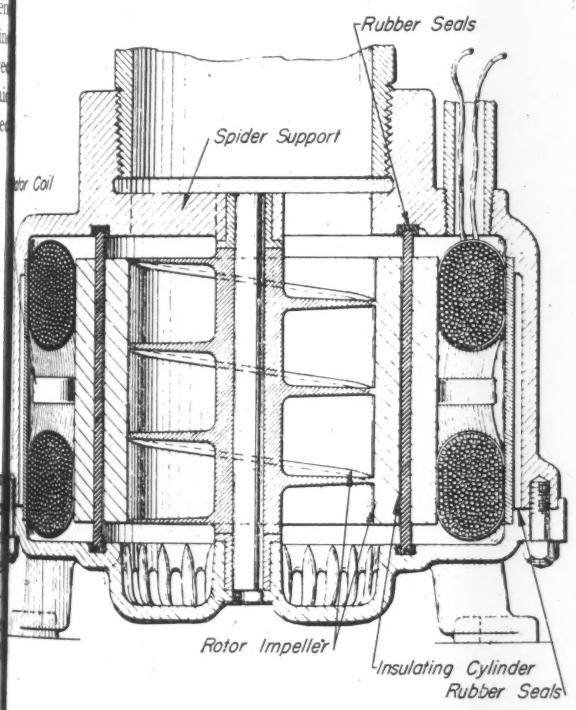
"Weldability"—as applied to the reaction of a base metal to the heat directly adjacent to a weld—can be predetermined and loss of ductility in the material next to a weld, usually caused by an excessive cooling rate which hardens and embrittles the steel, can be controlled. This book on the weldability of steels has been compiled with the intention of promoting a practical method for handling this particular problem. Use of an end-quench and a notch-bend test, as outlined, enables the design engineer to prescribe the proper current and welding speed to insure safe ductility in the heat-affected zones. Factors for all plate thicknesses up to two inches are given in the book for butt, fillet and lap joints. Tables show the proper preheat to use and indicate when preheat is required for good results. Numerous examples and drawings illustrate clearly the step-by-step procedure to follow in making use of this new system for selecting steels and predicting ductility in the heated weld zone.

For those interested in the advancement of safe welding technique in the manufacture of any form of weldment will find this a highly informative and interesting book.

Noteworthy Patents

Motor-Pump Uses No Direct Coupling

DAPTED to serve as a compact submersible sump pump, the basic design of the unit shown in the companying illustration can, with minor alterations, be d to fill a variety of special requirements as either fluid-impelling pump or a fluid meter. Covered in a es of patents granted to Albert R. Pezzillo, the device



his sump pump uses a special impeller fitted within the rotor of the electric drive motor

atures a pumping impeller fitted within the rotor ele- ent of a specially designed electric motor, affording raight-line flow of the pumped medium regardless of the rangement.

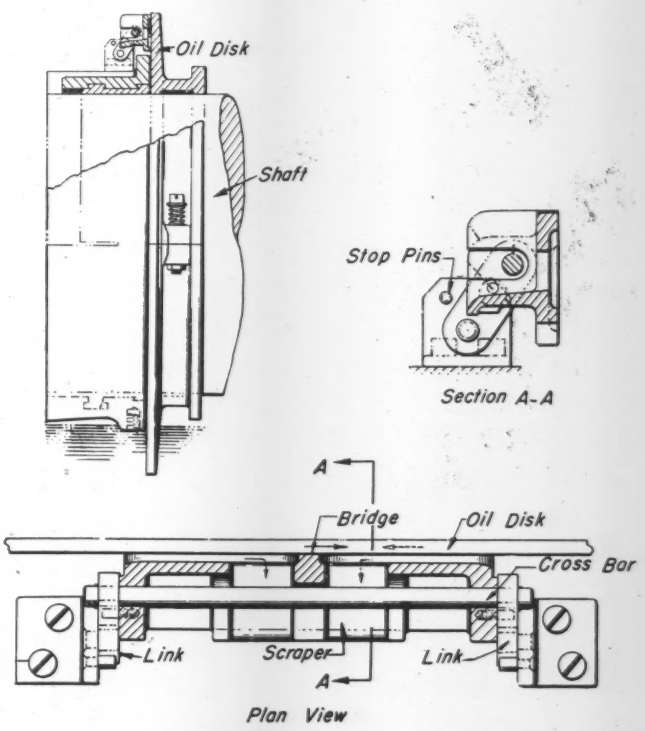
A continuous helical impeller blade is mounted on, or ade integral with, a tubular hub and fastened into a otor unit to form a single element. The hub of this otor-impeller is carried on a solid shaft which is retained y bearings fitted into spider sections that are integral ith the mating end-bell housings. Electric motor stator oils of standard design are hermetically sealed into the ell housings by means of a rubber-gasketed cylinder. his cylinder separates and electrically insulates the stator minations from the fluid being pumped. Design of the d bells is such that assembly automatically aligns and aintains the required uniform gap between the rotor- peller and the insulating cylinder.

By connecting the motor windings to a suitably calibrated meter the device illustrated may, in one of its many design variations, be used as a fluid flow meter. Slight alterations will also adapt the device to use in mixing fluids, aerating, etc.

Disk Oiler for High-Speed Journals

INHERENT difficulties and deficiencies usually found in ring-type journal oilers as well as those of present rotating-disk type oiling systems are said to be eliminated by the new design shown in the accompanying illustration. Patent 2,357,410, recently assigned to the Kingsbury Machine Works Inc., covers this simple effective means for supplying adequate quantities of oil to high-speed journals or other bearings.

Gravity is employed as the sole means of retaining the scraper contact with the disk face. Oil carried by the disk is removed by the scraping action of the edge of the cross bridge and deflected through one or more openings along a passage into the bearing reservoir. Pivotal mounting of the scraper unit provides a considerable latitude in mounting the oil disk relative to the bearing shell. The assembly is completely self-aligning and well adapted to variations in temperature or speed of operation as well as movements of the disk or shaft.



Gravity controls the self-aligning scraper unit in this compact disk-type journal lubricator

Dependability with Heaters

(*Concluded from Page* 117)

needs auxiliary heating, because the flexible heater can be distributed across the metal face of the magazine, resulting in a relatively low average surface temperature.

The direct-current drive motors for these magazines are also protected with a thermal cut-out. The motor circuit is opened within less than a minute's time should the mechanism become jammed. The locking of the motor armature causes the locked-armature current to pass through a small heater wire in close contact with a bimetal disk the thermal protector. The local heating of this d causes it to snap open, thereby opening the motor circu This protection is especially necessary on motors design to give a large shaft output for a short duty cycle of o to five minutes.

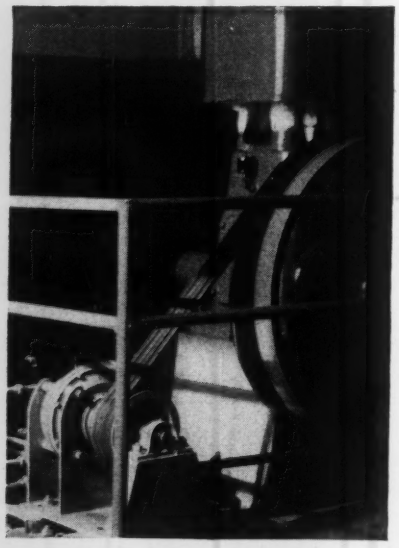
A similar application has been made for surface heati of aerial camera lens cones. The heater consists nichrome wire wound into a webbing of asbestos clo with the outer surface of the heater insulated by a wra ping of fiberglass cloth and the assembly impregnat with a flexible varnish so that the heater remains flexi and can be wrapped around circular surfaces.

Water-Brake Simplifies Engine Contro

By Earle S. Burnett
Mechanical Engineer
U. S. Bureau of Mines, Amarillo, Tex.

EXPANSION engines or, simply, expanders, have been used for many years for the production of intense cold through external work done by compressed air, nitrogen or other gas expanding behind the pistons. The chief application has been in plants for the liquefaction of air and the separation of its constituents, particularly oxygen, by methods originally developed early in this century by the Frenchman, Georges Claude.

Most expanders are small, single-acting, vertical engines of 5 to 20 horsepower, operating with fixed cut-off at nearly constant head pressure of about 600 pounds per square inch. They develop power and produce refrigeration in nearly direct proportion to their speed. This power is absorbed and their speed controlled in most installations of the past by electric generators, either direct - connected or belt-driven.



External work done by the gas expanding in these engines results in a cold exhaust which provides the desired refrigeration. Practically, this external work is so small in comparison with the total compressor load that efforts to recover and to apply it usefully have not been considered warranted. The electrical energy generated is usually dissipated as heat in air-cooled grid resistances.

Although electrical loading of the expanders provides an easy, flexible method of speed control, the necessary electrical equipment is expensive. If it is operated in s uations associated with liquefaction of combustible gas it must be isolated in chambers supplied with a plenum fresh air to safeguard against possible explosion hazar in atmospheres that might be contaminated by leakag

To avoid this expense and possible hazard in numero expander installations at plants recently constructed f extraction of helium from natural gas, belt-driven hydrau ic couplings and "hydrotarder" brakes were selected fro several devices considered, after experimental work by th author had established their suitability. Such an install tion is illustrated in the accompanying photograph.

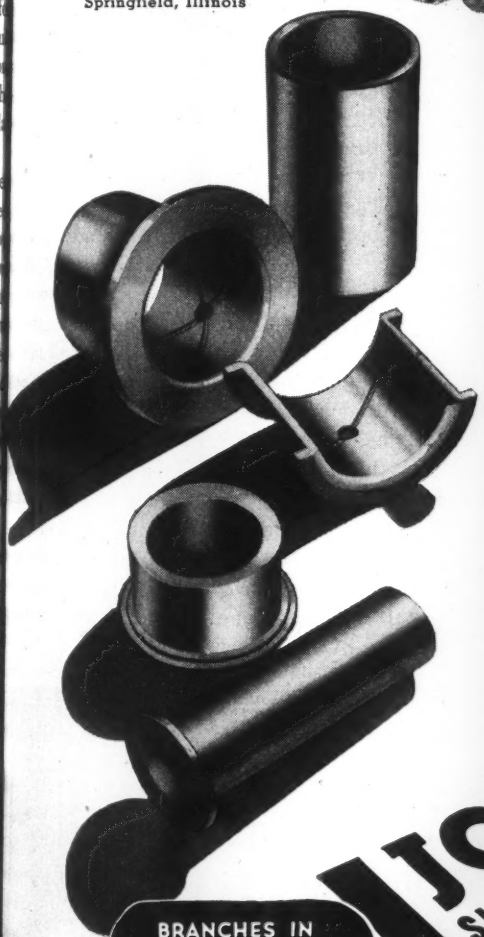
The brake is designed specifically for absorbing powe by a churning action of impellers on water circulate through the enclosing housing. The power absorbed b this action, for any one size of hydrotarder, depends upo the quantity of water within the housing, and varies abou as the cube of the speed of impeller rotation for a give filling of water. The nearly constant driving torque de veloped by an expander, stepped up in speed by multipl "sparkproof" V-belt flywheel drive to sheaves on the brak develops therein an opposing torque which quickly st bilizes at any desired speed determined by the quantit of water that the hydrotarder is permitted to retain with in the system.

Obviously, this water will be heated by the expande work and must be cooled. At the helium plants men tioned a closed system is used. The centrifugal pumpin action of the brake impellers is utilized to drive some o the water out through one pass of a small shell and tube type water cooler and back into the suction or supply con nection of the brake, thus automatically maintaining constant quantity of water within it.

Manually operated self-closing supply and bleed-o valves attached to this closed water circuit permit chang ing the total quantity of water within the brake housing Addition of water reduces the speed; withdrawal of wate increases the speed. Response to these manipulations i immediate and speed changes are established withou hunting. An expander started under full pressure wi not overrun in stabilizing. From the standpoint of speed control for this particular application, these arrangement and manipulations are simple, positive and satisfactory

TOUGH

Baker Hydraulic Bulldozer made by Baker Manufacturing Co. Springfield, Illinois



When an air strip is needed in a hurry . . . or a pillbox is to be demolished . . . the call is for a Bulldozer. In the present war these sturdy earth movers have earned their chevrons along with tanks, guns, planes and ships. Most fighting units class them as indispensable.

The Baker Bulldozer is a good example. In war and in peace they deserve their well earned reputation for being TOUGH. They do their work fast . . . efficiently . . . and keep right on doing it day in and day out. One reason why Baker Bulldozers deliver such excellent performance is that they are equipped with Johnson Sleeve Type Bearings.

Helping manufacturers of all types of equipment to build superior performance into their product is a welcome assignment for Johnson Bronze. When you need advice on any type of bearing application, consult with *Sleeve Bearing Headquarters.*

JOHNSON BRONZE CO.

525 S. MILL STREET NEW CASTLE, PA.

BRANCHES IN 18 INDUSTRIAL CENTERS

JOHNSON SLEEVE BEARING HEADQUARTERS BRONZE

Electronics

(*Continued from Page* 146)

the two-point control type *B*. The rheostat is operated by a reversing motor of the split-field series type, energized by other contacts on the relays of the two photoelectric relays. Thus, when the position of the loop moves out of either end of the limits set by the two light beams, there is a momentary change in speed due to the action of the photoelectric relay contacts on the small series field resistors and a slight change in the average speed as set by the motor-operated rheostat. (On types *B, C, D* and *E*, when light reaches the phototube holder, relays *CR* pick up, closing the normally open—*N.O.*—contacts and opening the normally closed—*N.C.*—contacts.)

TYPE F: All-electronic loop control usually consists of a photoelectric means of detecting loop position, an amplifier, and a thyratron rectifier circuit (see *Fig.* 7). The phototube receives a varying quantity of light, dependent on loop position, and produces a signal voltage which indicates the position of the loop. This signal, when amplified, controls the output voltage of the thyratron rectifier and hence the field current and speed of the motor. For a low position of the loop, the control produces a strong motor field and this decreases the motor speed, causing the loop to rise. For a high position of the loop, the control produces a weak motor field which of course

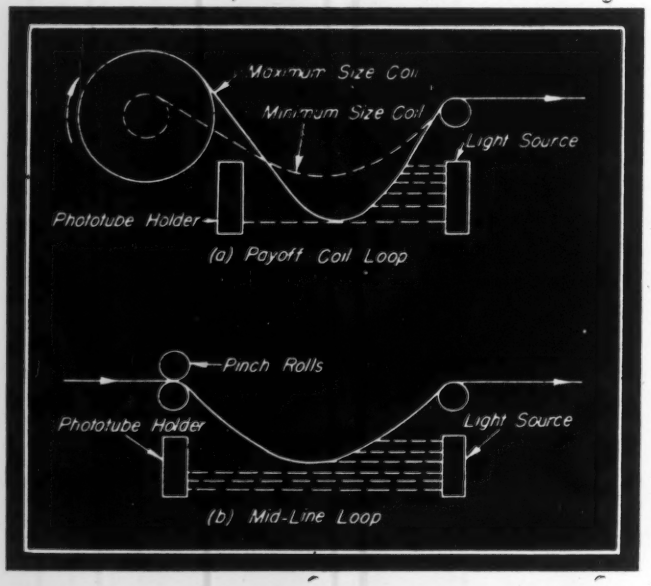


Fig. 8—Ideal positions for light sources and phototube holders to control payoff-coil and mid-line loops

increases the motor speed and causes the loop to lower. For each position of the loop in the control zone there is a corresponding definite motor speed. The loop will ride steadily in that position at which the resulting motor speed feeds material into the loop at the same linear speed as that at which it is withdrawn. As the coil diameter decreases on a payoff loop, the loop rises slowly but steadily to increase the speed of the motor. The control contains stabilizing circuits to compensate for time delays in the response of the elements of the system and thus prevents "hunting" (a continuous oscillation of the whole control system in which the motor speed and the loop position continuously changing).

It is possible to design the amplifier of an electronic lo control to furnish excitation for a motor-generator set, su as an amplidyne generator, which then would provide field voltage for the motor. Another possibility for all-electronic loop control is to use parallel high-vacu tubes of the power amplifier type connected in parallel in series with the field of the motor and control the mo speed by varying the current through these tubes. Ho ever, it is not practical to use high-vacuum tubes direc to control the field current on motors larger than about horsepower.

STABILITY OF AUTOMATIC LOOP CONTROLS: Type loop controls are intended for installations where stabil of operation is inherently achieved because of the relati ly wide control zone, narrow range of speed control a low material speed. Under such conditions the loop ri steadily in the position necessary to produce the requir material speed.

Basically, all of the automatic loop controls except typ *A* and *F* function by a cycling which is similar to huntin In types *B, C, D,* and *E,* the loop will change its positi until it passes beyond a definite limit. Then there is abrupt change to a second speed which moves the lo back from the limit. This is a repeating cycle that diffe only in the rate at which it runs out and the frequency correction. The second or correction speed should diff from the first speed by sufficient magnitude to permit t motor to coast slightly and move the loop back well b yond the limit position. A slight difference in these tw speeds would produce too frequent operation of the co trol and the loop would jiggle continuously at the limit p sition. In types *B, C* and *E,* the operating speeds at t two limits should be far enough apart to bound the norm variations in the operating speed of the material. In t type *F* control, the motor-operated rheostat corrects for slow changes in speed and thus greatly reduces the fr quency of cycling. It is desirable that the small field r sistors and the speed-control resistor for the rheostat mot be adjustable so that adjustments can be made to secu optimum results at the time of installation.

The two most important characteristics of the all-ele tronic loop control (type *F*) are stability and speed of r sponse. Stability or freedom from hunting is necessa to achieve smooth operation. Fast speed of response required to maintain the loop during rapid acceleratio of the line. Stability in an all-electronic system increas

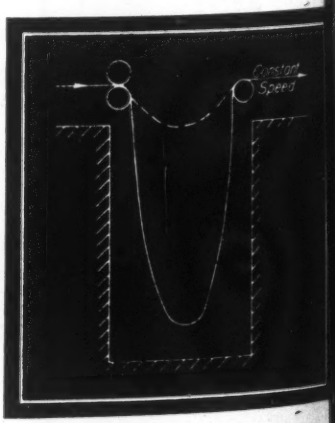


Fig. 9 — Deep-pit loop offers storage capacity

An Important Plastics Announcement BY DOW

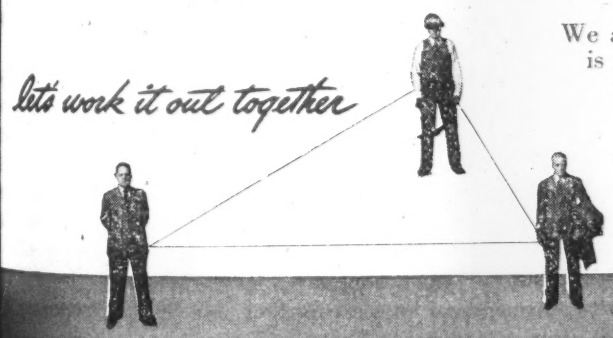
STYRALOY

A NEW PLASTIC FOR INDUSTRY

Highly significant among countless new materials developed for war requirements is Styraloy—trade-mark name for a remarkable group of plastics possessing properties intermediate between rubber-like materials and rigid thermoplastics.

"Working it out together" with the Armed Forces, Dow developed the first of this impressive new line—Styraloy 22—to provide a one-piece cable sheathing with a low power loss at high frequencies and possessing great durability and flexibility. These unique qualities—combined with others presented below in capsule form—point to its use in a broad range of products. As a result, unlike many war-born materials, Styraloy anticipates a peacetime career of great importance.

Now that Styraloy is available for commercial purposes, molders and manufacturers or designers will find Dow equally willing to cooperate with them in developing to the fullest extent the numerous applications indicated by the impressive list of Styraloy's properties. "Let's work it out together."



We at Dow know from experience that success in plastics is not a one-man nor even a one-industry job. It calls for the combined skill and cooperation of manufacturer or designer plus fabricator plus raw materials producer. Working together, this team saves time and money and puts plastics to work successfully. Call us—we'll do our part.

PRESENT AND POTENTIAL USES: One-piece cable sheathing; handles for tools, household appliances, etc.; gaskets; bushings; coil forms; floor mats; scuff plates; many applications still to be ascertained.

PROPERTIES AND ADVANTAGES: High dielectric strength, low power loss over all frequencies. Power factor only .005 at 100-300 megacycles. Flexible and shock resistant from −90° F. to 212° F. Specific gravity less than 1 (floats in water). Water absorption only .2 to .5%. Resists heat, ozone, and most chemicals. Highly resistant to abrasion. Resists permanent indentation. Ideally suited to extrusion of complex cross sections and readily fabricated by other molding techniques. Easily machined.

THE DOW CHEMICAL COMPANY · MIDLAND, MICHIGAN

New York • Boston • Philadelphia • Washington • Cleveland • Detroit
Chicago • St. Louis • Houston • San Francisco • Los Angeles • Seattle

DOW

PLASTICS

STYRON • ETHOCEL • ETHOCEL SHEETING
SARAN • SARAN FILM • STRIPCOAT • STYRALOY

TABLE I

Applications of Automatic Loop Controls

Type of Loop	Material Speed (ft. per min.)	Motor Speed Range Required	Change in Inertia Load on Motor°	Control Zone Width (inches)	Type of Automatic Loop Control A Loop Rheostat	B One Point	C Two Point	D Rider-Roll Rheostat	E Motor-Operated Rheostat	F All-Electr
Payoff	0-100	5:1	5:1	20	X	...	...	X	X	X
Payoff	100-1000	5:1	50:1	20	...	...	...		X	X
Payoff	100-1000	where 5:1 is obtained by rider-roll rheostat with ± 7% additional by two-point control				...	...	X	...	...
		5:1	50:1	20		...	...		...	...
Mid-line	0-100	±10%	1:1	2	X	X	X	...	X	X
Mid-line	100-1000	±10%	1:1	20	X	?	X	...	X	X
Mid-line	1000-3000	±10%	1:1	20	...	...	...	...	X	X
Mid-line	0-100	±30%	1:1	5	X	...	X	...	X	X
Mid-line	100-1000	±30%	1:1	20	...	...	?	...	X	X
Storage	0-100	+50 −10%	1:1	5	X	?	X	...	X	X
Storage	100-1000	+50 −10%	1:1	20	...	...	X	...	X	X

Notes: X indicates control is generally suitable for conditions indicated.
? indicates control is questionable for conditions indicated, especially at the highest speed.
° Expressed as a ratio or range of the total inertias involved, including motor inertia.

directly with the width of the modulating zone; that is, the control zone in which each position of the loop corresponds to a different motor speed. Stability decreases when the speed of the material increases and when the range of speed required to hold the loop increases. Inertia of the motor and its connected load also affect stability. A low ratio of the friction load to inertia load on a motor decreases the stability of the system. High speed of response and stability of all-electronic loop control are due solely to the proper use of stabilizing or anti-hunt elec-

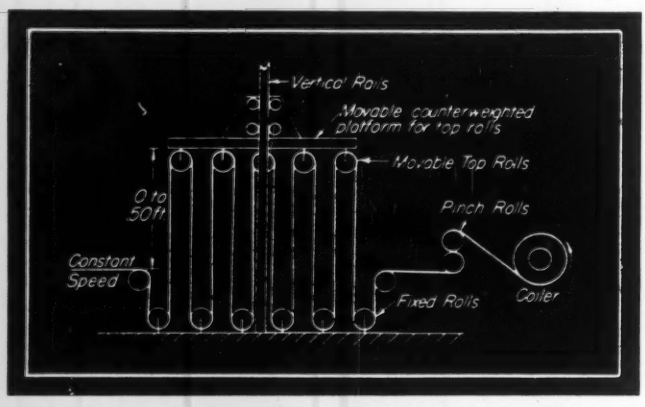


Fig. 10—Looping tower is employed where considerable storage capacity is required

trical circuits as combined with an adequate width of control zone.

There are two basic electrical methods of achieving stability in all electronic control circuits. The first—the "slow down" method—consists of decreasing the speed of response of certain elements of the system until stable operation is obtained. The second—the "speed up" method—increases the speed of response of the control. The second method anticipates time delay and adds to the correction signal an antihunt signal which is a function of the rate of change of the correction signal. This "speed-up" antihunt signal is comparable to the action of an experienced coil handling operator who anticipates the time delay and moves the field rheostat to decrease the rate of change of the loop position before the rapidly changing loop reaches the desired position.

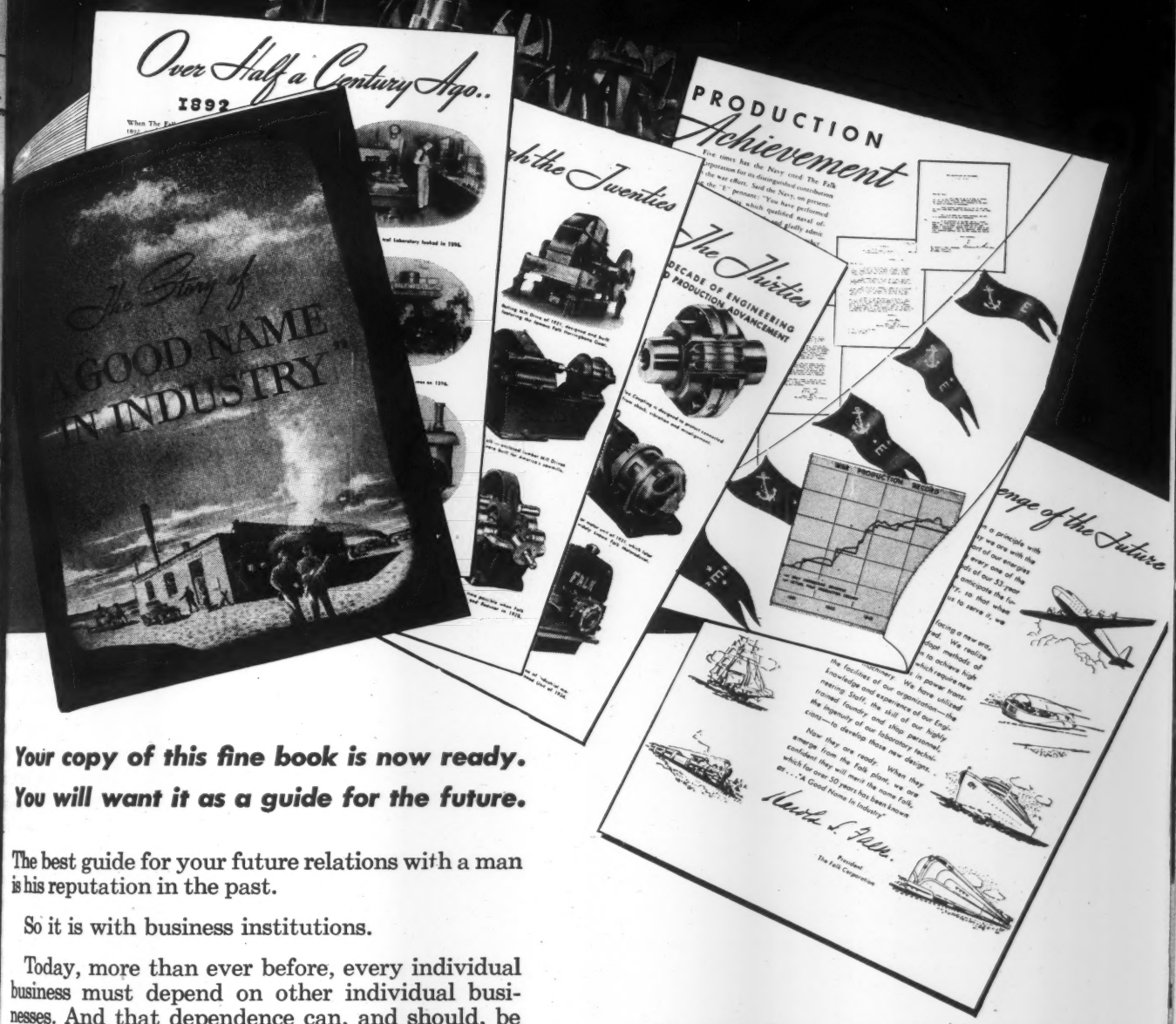
LOCATION OF PHOTOELECTRIC DETECTING UNITS: Light sources and phototube holders generally can be located out of the way where they cannot be easily damaged and do not interfere with the threading of the line. Depth the loop should be at least one-fourth and preferably on third or more of the length. The ideal position of t light source and phototube holder is one in which t beam of light is horizontal and parallel to the length the line and passes directly under the center of the botto of the loop. Good locations for these units are shown *Fig.* 8.

It is difficult and often impossible to mount the lig source and phototube holder in the ideal position, partic larly on payoff loops and extremely shallow loops. The either of the units can be placed at one side of the loo with the other under the loop as shown in *Fig.* 1. Whe the loop is very low, the beam of light should not pa over the loop and give a reverse directional signal. Var ing widths of material add further complications when th ideal positions cannot be used. When the beam of lig is not horizontal or passes under the side edge and not th bottom of the loop, the loop may develop a slight swing o payoff loops of high-speed lines where fast-response phot electric loop controls are used. These swings change th amount of light reaching the phototube holder, even thoug the length of the material in the loop does not change ap preciably. Such light changes produce small variations i the speed of the payoff coil, which serve as impulses t sustain the swing of the loop.

For loops having considerable storage capacity, eithe the deep-pit loop of *Fig.* 9 or the looping tower of *Fig.* 1 can be used. Loops at the exit ends of lines normally ru empty—that is, with little material in them—but fill u when the coiler stops. When the coiler starts up again the loop control automatically runs the coiler at a speed i excess of the line speed to return the loop to its norma position. Conditions are similar for storage loops at th entry ends of lines except that here the loops normally ru full of material.

SELECTING THE PROPER TYPE OF CONTROL: Range application of each type of automatic loop control shown in TABLE I. Since this table lists two or more type of control for each application, one type must be selecte on the basis of cost, desired operating characteristics, an expected maintenance. For low-speed lines it is easy t apply properly the simple automatic loop controls of type *A, B* and *C*. But for high-speed lines and the more com plicated controls of types *D, E* and *F*, it is recommende that the application engineering section of the manufa turers of the apparatus be consulted.

HE PAST FORETELLS THE FUTURE



Your copy of this fine book is now ready. You will want it as a guide for the future.

The best guide for your future relations with a man is his reputation in the past.

So it is with business institutions.

Today, more than ever before, every individual business must depend on other individual businesses. And that dependence can, and should, be placed only upon those business institutions whose record of performance as to policy, product, service and general attitude has been flawless in the past.

Many new businesses exist today. Many others, old in years, have not kept young in thinking . . . and are passing from the national picture. This has been the history of business since the world began.

Yet, through all the economic changes, certain institutions have grown old in years, but have kept young in spirit and have constantly improved upon their original reason for existence.

The book illustrated above deals with the history of The Falk Corporation, its ideals, accomplishments, and plans for the future so far as those plans can now be projected.

You, too, in your effort to render a service to your industry, must of necessity have followed many of the principles set forth in this book. For that reason we believe that the book will interest you and will provide you with reasons why The Falk Corporation will be a sound source of supply for the machinery you will need to shave pennies from production costs and improve your product, in order to compete in the salesmen's war to come.

THE FALK CORPORATION, *Milwaukee 8 Wisconsin*

ALWAYS PAYS TO CONSULT **FALK** . . . A GOOD NAME IN INDUSTRY

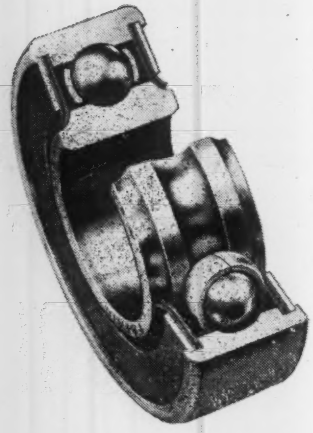
For over fifty years Precision Manufacturers of

Speed Reducers • Motoreducers • Flexible Couplings • Herringbone and Single Helical Gears • Heavy Gear Drives
Marine and Diesel Engine Gear Drives and Clutches • Steel Castings • Contract Welding and Machine Work.

District offices, representatives, or distributors in principal cities.

New PARTS AND MATERIALS

Bearings Offer Positive Sealing



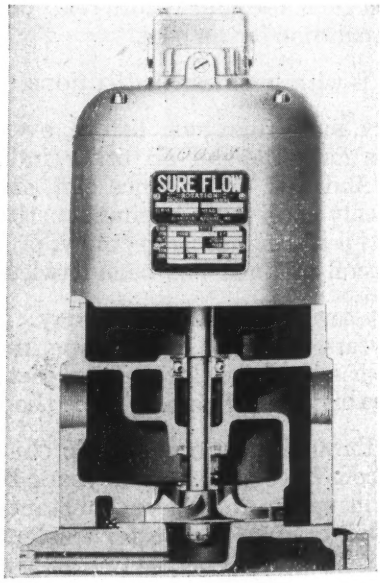
BEING OFFERED by the Fafnir Bearing Co., New Britain, Conn., is a new type of sealed ball bearing known as "Plya-Seal." This sealing element consists of a diaphragm type contact seal comprising two members —a flat, flexible sealing washer of synthetic rubber-impregnated fabric and a split retaining ring of spring steel. Due to the minimum space required for the two seal parts, the bearings are held to the same widths as standard unsealed bearings. The sealing washer, held in the outer ring, does not rotate with the inner ring but is in contact with a ground groove to form an effective seal with a minimum of friction. Easily removed and replaced to allow inspection, washing and regreasing, the bearing is especially suitable for applications where periodic overhauls of equipment are standard practice. Maximum retention of lubricant and exclusion of dirt and liquids are also assured. The seal causes no distortion of the outer ring or race, and is noncapillary and impervious to grease, oil, gasoline, water and a variety of solvents. It resists heat to a high degree and is unaffected by subzero temperatures as well. According to the company, the bearings have been performance-proved in military equipment, and are approved for aircraft by the Army Air Forces and the Bureau of Aeronautics, U. S. Navy. They have also been used successfully on many types of industrial machinery.

Heavy-Duty Slip Clutch

ANNOUNCED BY The Hilliard Corp., 103 West Fourth street, Elmira, N. Y., is a heavy-duty slip clutch which is designed to transmit power between a shaft and a gear, sprocket, sheave, or pulley mounted on the extended sleeve. Bolted to the extended sleeve, the outer driving ring engages the lined plate by gear-tooth splines. Asbestos linings are riveted to each side of the lined plate, and these linings bear against the friction surfaces of the hub flange and pressure plate. Pressure on the friction surfaces is obtained by a series of coil springs placed between the pressure plate, mounted on hub with gear tooth splines, and a collar carried on the gear tooth splines of the hub. An adjusting nut, threaded over the end of the splines, provides for regulation of the spring pressure and det mines the torque setting at which the clutch will slip. positive locking position for the adjusting nut is provid by each individual spline on the hub, and the number splines used permits close torque adjustment. The u can be furnished in a variety of designs, including cou lings, in both single and multiple plate types.

Centrifugal Coolant Pumps

INCLUDING twenty-seven different sizes and types a designated as "Logan Sure-Flow" pumps, an improved li of centrifugal pumps has been announced by Logansp Machine Co. Inc., Logansport, Ind. Designed especial for pumping coolants and cutting oils, they are adaptab for circulating cooling liquids, pumping or circulati water or quenching oil and general liquid transfer servic One of the outstanding features of the pumps is the op impeller, eliminating need for close clearances or metal-t metal contact of running parts. This allows pumping liquids containing some abrasives, filings and other forei matter without damage to the pump. Equipped with integral motor drive, the pump is available in foot-mou



ed, bracket-mounted, flange-mounted, grinder and su merged types. Self-priming without submerging, in th base and bracket-mounted models, affords latitude in l cating the pump in relation to the liquid to be pumpe The totally enclosed ball-bearing motor in the base, brack and flange-mounted types is protected by a rotary se against splashing liquids, abrasive dust and other harm ful foreign elements. The grinder and submerged mode

OTISCOLOY
HIGH TENSILE STEEL SHEETS

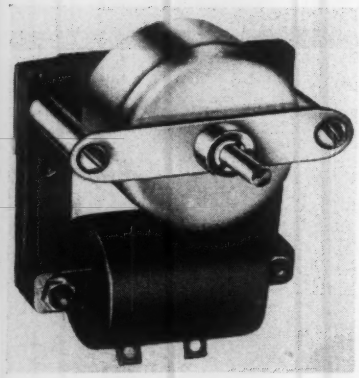
J&L STEEL

A high strength steel that is easily fabricated and readily welded. May be hot or cold formed. Affords reductions in weight . . . its greater strength permits use of lighter gauges. Resists corrosion. Otiscoloy available in both sheets and plates for a wide variety of applications.

JONES & LAUGHLIN STEEL CORPORATION
PITTSBURGH 30, PENNSYLVANIA

have no seals, an overflow hole in the top of the column preventing liquid within the pump from reaching the motor bearings. Since maximum power is consumed at zero pressure, the pumps cannot be overloaded and any increase in static head, friction head or back pressure reduces delivery and power requirements.

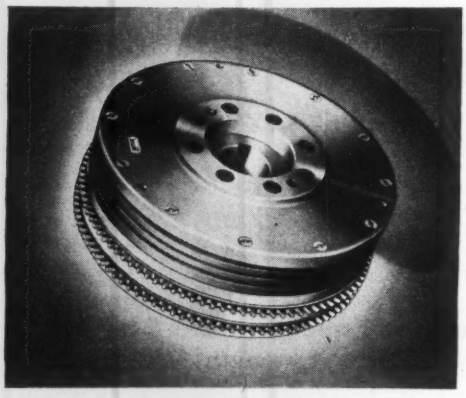
Timing and Control Motor



DEVELOPED BY Warren Telechron Co., Ashland, Mass., a new timing and control motor with terminal shaft speed of one revolution per hour has an input rating of 2 watts and can be furnished in other speeds up to and including six revolutions per hour. A special oil gland has been built into the terminal shaft bearing, extending its life under industrial and domestic operating conditions. Terminal shaft is concentric with the outer case of the motor. A companion model, having a terminal shaft speed of three revolutions per hour, has also been developed. This design has an eccentric terminal shaft placed normally in the 12 o'clock position in respect to the outer case. It also can be furnished at speeds of from 3 to 6 revolutions per hour.

Air-Actuated Clutch

DESIGNATED AS Model P, a new clutch announced by Twin Disc Clutch Co., Racine, Wis., can be used wherever the company's Model E (friction) clutch has application. The new clutch is particularly advantageous where operation by remote control is desired. In many features of design and construction Model P is similar to Model E, having a hub and back plate, center plate, floating plate and friction disks of essentially the same proportions.



Model P design provides two different methods for h dling the actuating air supply: (1) By drilling the sh and using an end shaft rotary seal; and (2) by usin special shaft-around, or midshaft rotary air seal. T first is used where clutch is mounted near end of sh while the latter where the clutches are mounted in drums, several feet from closest shaft end. The only m ing parts of the actuating mechanism, other than spri are the piston, piston seals and floating plate. Piston s are woven asbestos and so located as to be well remo from heat induced by friction. The Model P clutch furnished in a wide range of sizes (14 to 36 inches) capacities (65 to 895 horsepower).

Snap-Action Pushbuttons

KNOWN AS the Snapit switch a small momentary pushbutton switch has been announced by Grayhill, One North Pulaski road, Chicago 24. The round phenolic body of the switch measures ⅞-inch in diameter and 1⅞ inches in height. It is mounted on a ⅜-32 bushing, 7/16-inch long, and held by two mounting nuts. The fixed contacts are fine silver overlay on phosphor bronze, threaded and held in place. Electrical connection is made by brass screws. The contact gap is .040-inch on each contact, making the to contact gap which breaks the circuit .080-inch. T Snapit switch employs a snap-action principle which a sures fast make and break as well as a contact pressure approximately 35 grams, making the switch practical f use on direct current. It operates on a .0625-inch mov ment of the pushbutton and carries a current rating of amperes at 115 volts alternating current, and 2 amperes 115 volts direct current. According to the company, it general practice on direct current to increase the ratin with a decrease in the voltage. A red pushbutton ind cates the normally-open, single-pole switch, while a blac pushbutton, the normally-closed, single-pole type.

Electronic Cycle Counter

DESIGNED FOR industrial and laboratory uses, a ne two-decade electronic counter of The Potter Instrume Co., 136-56 Roosevelt avenue, Flushing, N. Y., can als be used for counting and calibrating the actual number cycles that resistance welding timers apply. In additio it can be used as an interval timer by connecting it throug a switch to a known external frequency. When switc is closed and opened, the unit will count the number cycles of the known frequency that have passed in th closed-switched time interval, giving a reading in terms the number of cycles of the known frequency. The 6

New leather packings, made a new way

VIM resin-impregnated "V" packings have these merits:

They are molded to shape, so that the grain, or smoother side of the leather, will contact the cylinder or plunger. This reduces friction, as compared to a trimmed O.D. and I.D. packing whereby the flesh side contacts the operating surface.

But that is only one advantage of VIM Resin-Impregnated "V" Packings, which were originally developed for aircraft hydraulic use.

Its impregnation treatment renders it insoluble in, and unaffected by, any type of oil medium. This filler will not wash out.

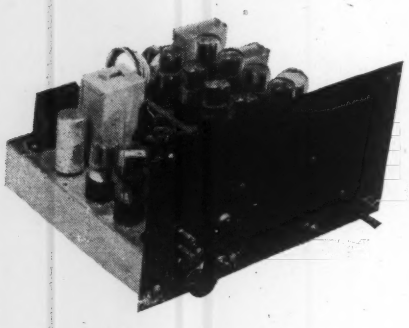
These packings are usable at temperatures between minus 65° and 180° F., at pressures from zero to as high as 60,000 PSI.

They're long-lived, and are giving excellent service on machine tools, presses, pumps, and other hydraulically operated equipment. For design data, write—

E. F. HOUGHTON & CO.
303 W. Lehigh Ave., Philadelphia 33, Pa.
CHICAGO SAN FRANCISCO DETROIT

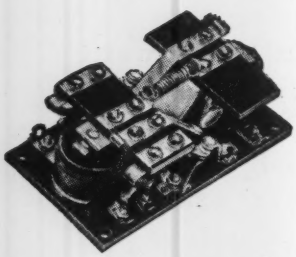
HOUGHTON'S *Engineered* **VIM** *Leather Packings*

cycle line may be used as the known frequency. The counter is actuated by a closing contact, sine wave, or pulse input, as from a photo-tube, at rates up to 1000 cycles per second. Each decade, dividing by ten, gives a scaling factor of 100. The count for 0 to 99 appears on two banks of neon lamps. Connected to the counter output is a telephone-type relay, and its output terminal con-



tacts close once for each 100 input cycles. A conventional electromechanical counter may be connected to the output terminals to extend the count to as many places as desired. An important application of the unit is in its counting rates exceeding 10 cycles a second. Using a complement of 11 tubes the counter can be supplied with switches to make it predetermining. Operation is from a 60 cycle, 105 to 125 volt line. Weight is 25 pounds.

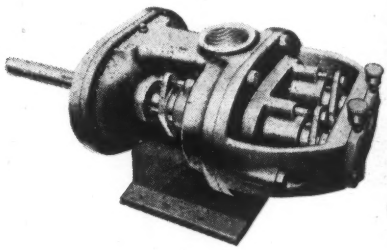
Latch Interlock Relays



CONSTRUCTED to aviation specifications Series 50XBX 2-coil relays have been added to the line of "Memory" relays produced by Struthers-Dunn Inc., 1321 Arch street, Philadelphia 7. The latch-in relay construction consists of a new style positive interlock between the two symmetrical operating elements and requires no extraneous parts other than integral extensions of the coil "armatures" themselves. It operates from a momentary impulse, and a minimum of power. Application of power to one coil latches the contacts into one position. Power is then applied to the other coil which throws the contacts into a latched-in second position. A third "unlatched" position, valuable for certain applications, can be obtained by energizing both coils simultaneously. This new design makes it easy to obtain make-before-break, or break-before-make contact combinations. Contacts do not interrupt the coil circuit until the "throw" is completed and contacts are locked in the new position. Relays of this type are produced in ratings from 6 to 200 amperes or more, and with any desired contact arrangement. Standard types provide for two auxiliary contacts, one in each coil circuit. Use of auxiliary contacts makes it possible to obtain operation over a wide range of voltages, alternating or direct current. The new relays are small and compact in design, and Type 50XBX103, for instance, double-pole, double-throw main contacts and is rated 6 amperes at 24 volts direct-current. Overall dimensions are 3 7/16 x 1 7/8 x 1 5/8 inches.

Positive Displacement Pumps

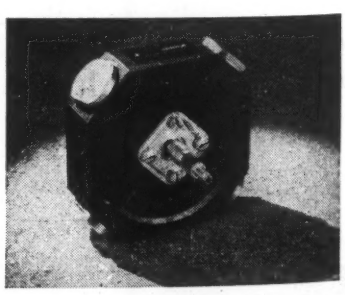
SELF-PRIMING under high vacuums or against high pressures, new positive displacement pumps of The Blackmer Pump Co., La Crosse, Wis., deliver a constant volume per revolution, not affected by speed or pressure. In operation the liquid is drawn into the intake side of the pump by a vacuum suction and expelled at the outlet side through piston action. Design is such that intake and outlet ports are open at all times, with a continuous seal between the two rotors, preventing liquid from returning into the intake chambers. These industrial pumps are available



capacities for ½ to 400 gallons per minute. Liquids with temperatures up to 210 degrees can be pumped successfully. The models are furnished for constant volume, variable capacities and automatically controlled volume in V-belt drive, direct motor drive, variable speed and tight or loose pulley types.

Hydraulic Selector Valve

DESIGNATED AS AN6213-1, an improved type of hydraulic four-way selector valve in which pressure drop has been reduced by half has been announced by the Pacific Division, Bendix Aviation Corp., 11610 Sherman Way, North Hollywood, Calif. Available for ⅜-inch tube size, the valve is radial in design and utilizes plastic poppet



valve elements. The body is a permanent mold aluminum casting and the cored fluid passages are oval, instead of round in cross section. This development coupled with the short distance traversed by the fluid through the valve, results in a pressure drop of only

EASTERN STAINLESS
SHEETS
CONTENTS
Page
Technical Data Section
Explanation of "Why Stainless Steel Resists Corrosion" . . 3
List of Eastern Stainless Grades and Types 7
Description of Eastern Stainless Grades with Illustrations of Typical Applications 8-28
Description of Eastern Stainless Sheet Sizes and Tolerances 29
Description of Eastern Stainless Finishes 30-31
General Data on Processing and Fabricating 32-35
Tables of Physical Properties 36-37
General Data on Corrosion 38-42
Appendix, General Reference Tables 38-42
Technical Service
The EASTERN STAINLESS technical staff is readily available to analyze existing or contemplated applications of stainless steel sheets. Its many years of experience in this special field and modern research and development facilities serve two major purposes. First, to aid in the development, establishment and scientific control of modern production methods. Second, to form a link between our production units and the consuming industries. This service relates to all phases of the production, processing and fabrication of stainless steels and thus is of special value to producer and consumer alike. Consult with our technical staff in fabricating matters, performance problems or new developments. EASTERN STAINLESS Technical Service will gladly respond with up-to-date information and advice.

New 1945 EASTERN STAINLESS *Catalog* will soon be ready!

Your free copy of the comprehensive new 80-page Eastern Stainless catalog will shortly be available. Compiled and edited by Eastern Stainless technical experts, it authoritatively covers modern applications of Stainless Steels in many great industries and institutions including your own.

Complete in every detail, profuse with interesting illustrations, the new 1945 Eastern Stainless catalog will be a valuable addition to your reference files . . . it'll serve you well as an office-handy encyclopedia giving authentic information on how Stainless Steels are best suited to your needs. Today—fill out and mail the coupon at the right. You will be sure to receive your free copy of the 1945 Eastern Stainless catalog as soon as it is ready.

EASTERN STAINLESS STEEL CORPORATION
Baltimore, Maryland Dept. 64

Gentlemen: Please reserve for me a copy of your free 1945 catalog "EASTERN STAINLESS STEEL SHEETS."

Firm name .....

Address .....

City .....State .....

Your name and title.....

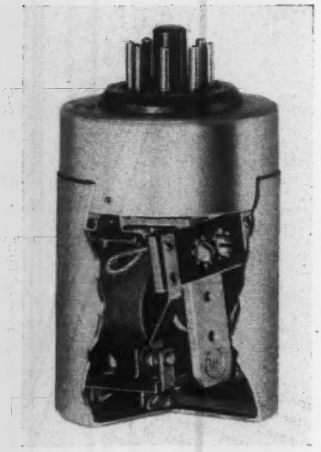
JML Co. B-C1

EASTERN STAINLESS STEEL CORPORATION
BALTIMORE, MARYLAND

SALES REPRESENTATIVES: CHICAGO, 201 N. Wells St., Phone: State 6533 • CLEVELAND, 1010 Euclid Avenue, Prospect 2020 DALLAS, 4533 Livingston Ave., Justin 8-5772 • DETROIT, 6505 Second Blvd., Madison 8570 • DETROIT, 906 Fisher Bldg., Trinity 11465 • LOS ANGELES, 10887 Chalon Road, Arizona 3-8273 • NEWARK, N. J., 972 Broad Street, Market 2-2068 PHILADELPHIA, 7444 Fayette St., Livingston 2002

pounds per square inch at 3½ gallons per minute. In addition to the ⅜-inch tube size valve, ½ and ¾-inch tube size valves are also available.

Plug-in Encased Relay

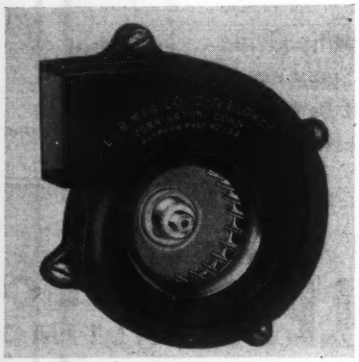


ENCLOSED IN a metal can and fitted with a standard octal plug base, a new plug-in relay has been introduced by Ward Leonard Electric Co., 31 South street, Mount Vernon, N. Y. Encased in a cylindrical metal housing 2 1/16 inches in diameter and 3⅛ inches high, the relay is rigidly supported against shock by means of a key in the center of an insulating disk that fits in the top of the case. It is made to operate on standard voltages up to 115 volts alternating or direct current, and is a modification of a popular type of unit used in small radio transmitters, aircraft control circuits, and similar applications where space is limited. Double-pole, double-throw contacts are rated 4 amperes at 115 volts, 60 cycle alternating current and at 24 volts direct current. From 25 to 115 volts direct current the rating is ½-ampere.

Hose for Oil and Coolant Lines

ESPECIALLY designed for connections on oil and coolant lines on Army plane engines, a new type of hose offered by the Mechanical Goods division, Goodyear Tire & Rubber Co., Akron, O., is compounded to withstand severe temperatures and has high strength and flexibility. The hose will withstand heat and will not harden or break under rigorous conditions. Produced in eleven sizes, the hose has been flight tested and approved by the Army Air Forces.

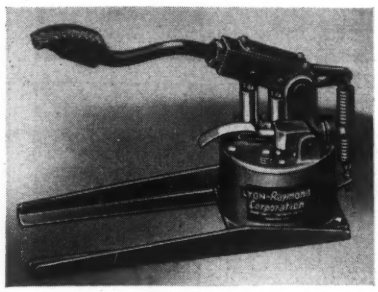
Lightweight Blower



OPERATING UNDER all conditions of climate and temperature, a new lightweight blower for heat dispersion, known as Model No. 2½, has been added to the line of L. R. Mfg. Co., Division of the Ripley Co., Torrington, Conn. With the addition of this new model, the company now has available blowers ranging in size from 3 to 6½ inches with weights from 2 ounces to 12 ounces, and with capacities ranging from 15 to [illegible] cubic feet per minute at 8000 revolutions per minute. The new unit weighs 3½ ounces and delivers 50 cubic feet per minute at 8000 revolutions per minute. It is available with shaft bores of either .1895 inch or ¼-inch. Housing is lightweight, high impact phenolic plastic. Available for either clockwise or counter clockwise rotation, the wheels are turbo-type cadmium plated steel.

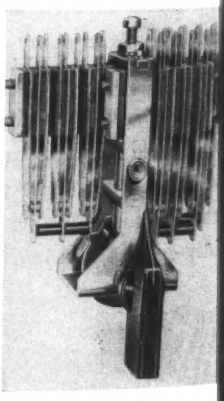
Hydraulic Foot Pump

FOR PRESSURES up to 10,000 pounds per square inch a two-speed foot pump has been developed by Lyon-Raymond Corp., 2024 Madison street, Greene, N. Y. The outstanding advantage of the pump is the automatic change speed feature. The pump has two pistons—one a high-speed, low-pressure piston, the other a small, slow speed, high-pressure piston. Up to 1000 pounds pressure



pistons pump oil, but at 1000 pounds the high-speed, low-pressure piston automatically cuts out and higher pressures (up to 10,000 pounds) are easily developed by the small high pressure piston. The pump is self-contained, designed for continuous, heavy-duty performance. It can be furnished with a base suitable for mounting to the machine or with an auxiliary base allowing it to be used on the floor.

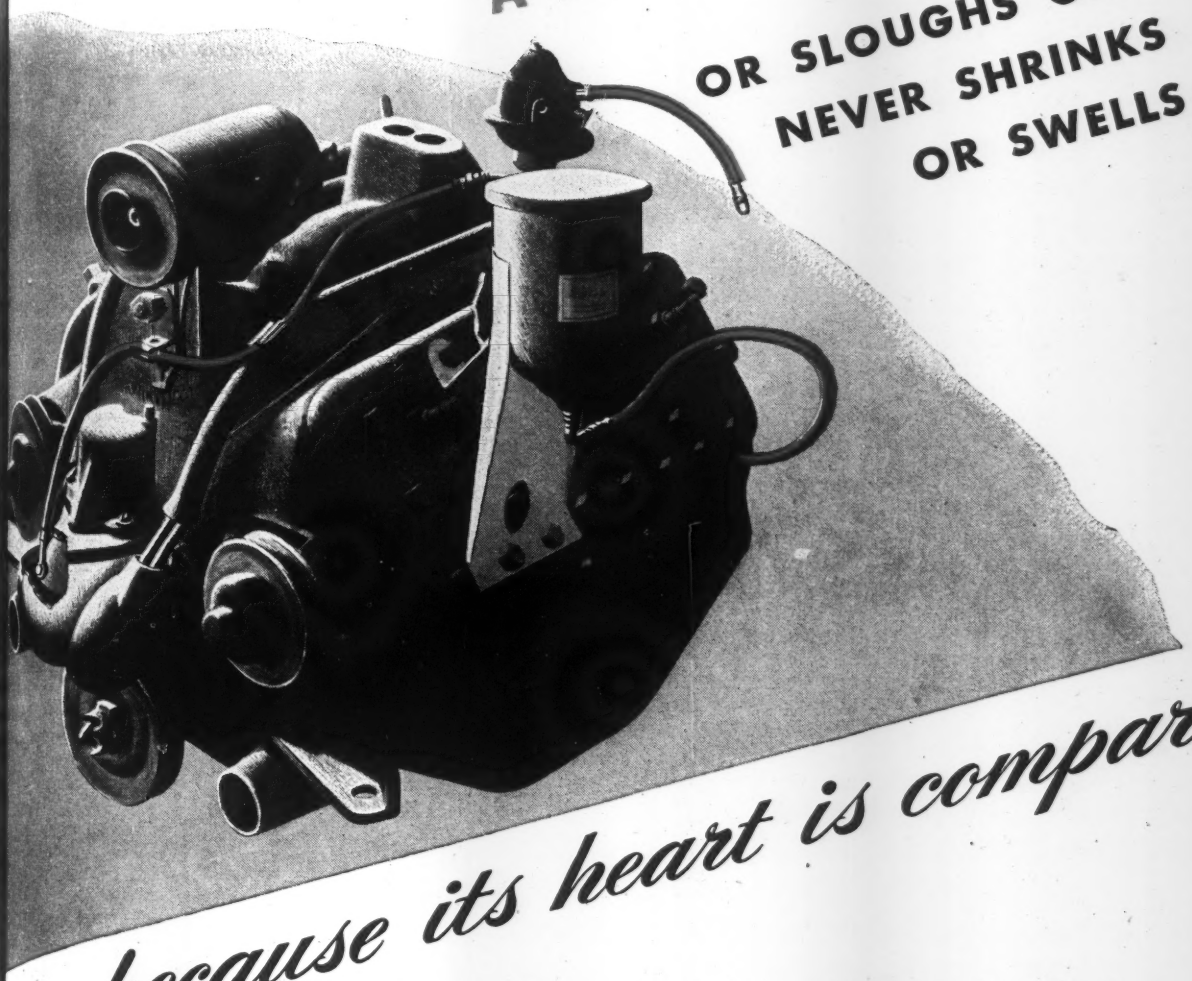
Switches for Panel Mounting



FOR PANEL MOUNTING a new switch — Series 6300—meets the need for a heavy-duty, multiple circuit turn switch. Available from Donald P. Mossman Inc., 612 North Michigan avenue, Chicago 11, as both a two position or three position switch, either locking or nonlocking, the new switch is light in weight and compact in size. The switch illustrated is Model 6300. Another series, No. 4200, also announced by the company recently, is a lever switch. Because of flexibility of circuit arrangements and unlimited combinations of contact forms, this series is proving itself for use in communication systems, testing apparatus, instruments, radio and other types of equipment.

THE TUBE IS THE HEART OF THE HOSE—INSIST UPON ITS BEING MADE OF COMPAR

A RESISTOFLEX FUEL LINE NEVER ROTS OR SLOUGHS OFF, NEVER SHRINKS OR SWELLS



... because its heart is compar

No matter what type of fuel is conveyed through it—heavy Diesel or high-octane aviation — the compar tube in Resistoflex flexible hose remains unchanged even after years of constant service.

Compar is immune to ALL fuels and oils. It cannot shrink, swell, rot, or slough off.

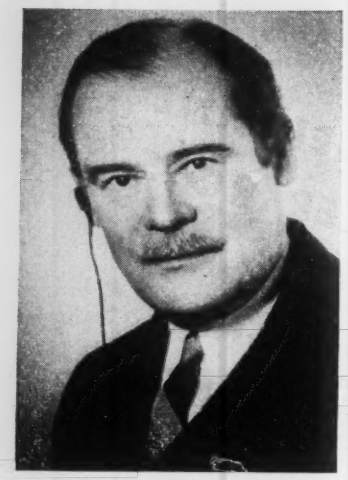
Integrally bonded fabric reenforcement—or woven wire covering—gives Resistoflex hose peak resistance to external abrasion, moisture, oil and general manhandling.

Inquire today about Resistoflex gasoline and oil lines. They are made in various constructions and formulations depending upon the specific job they are to do.

ARMY E NAVY

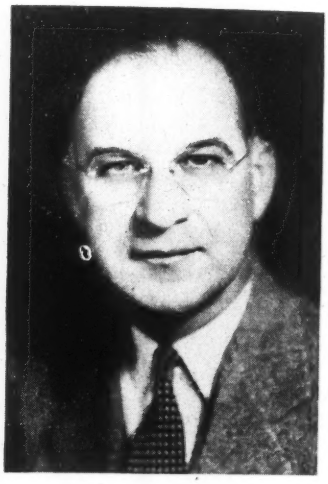
RESISTOFLEX

RESISTOFLEX CORPORATION, BELLEVILLE 9, N. J.



E. F. W. Alexanderson

A. W. LeFevre



J. M. Crawford

MEN... of machines

ERNST F. W. ALEXANDERSON, consulting engineer and radio expert at General Electric Co., recently was the recipient of the Edison Medal for 1944, awarded by the American Institute of Electrical Engineers and the Institute of Radio Engineers. Presentation was made in recognition of his "outstanding inventions and developments in radio, transportation, marine and power fields." Dr. Alexanderson is credited with 300 patents, taken out in his name in the foregoing fields. Born in Sweden, he is an electrical and mechanical engineer. Upon reading "Alternating Current Phenomena," by Dr. Charles P. Steinmetz, he decided to go to America to seek work with Dr. Steinmetz. He arrived in 1901 and visited him, and later at his recommendation obtained a drafting job with General Electric. After taking the company's engineering course he became a member of the engineering staff in 1904, designing generators under Dr. Steinmetz. Since then he has contributed largely to the development and improvement of alternators, radio and television equipment, motors, generators, electric ship propulsion and electric traction apparatus.

ARDEN W. LEFEVRE, new vice president of Stewart-Warner Corp. and director of engineering for the Alemite instrument and radio divisions, brings to his post a background of experience in automotive and related fields. This background—20 years of development and production engineering—was all obtained within the engineering organization he now heads. Mr. LeFevre joined Stewart-Warner as a designer in 1925. He specialized on th development of lubricating devices an also played a key role in the develo ment of a gasoline-burning heater, ori inally developed for automobiles b now adapted to aircraft and mari heating, and heating of tents, huts a trailers. Prior to his recent appointme he was general manager of Stewar Warner's Indianapolis plant. Previous he had been chief engineer for Alemi at Chicago.

JAMES MARK CRAWFORD, recen ly elected president of the Society Automotive Engineers for 1945, h been an automotive engineer through out his entire business life. He first ob tained industry-wide recognition in th 20's as chief engineer of a small car com pany and for the last fifteen years con tinued in design and has carried the to

FORT PITT *Alloy Steel Castings*

Guaranteed Analysis

When you buy FORT PITT Alloy Steel Castings, you can be certain that the analysis of the delivered job will be precisely as specified. Fort Pitt *quality control* follows every step of the production of your order, from the alloying of the steel to the final inspection of the finished casting. Every heat is sampled by skilled metallurgists and subjected to a series of exacting chemical and physical tests. Accurate duplication of analysis throughout the job is a "must" at FORT PITT foundry.

FORT PITT STEEL CASTING CO.

Div. of H. K. PORTER COMPANY, Inc.

Pittsburgh 22, Pennsylvania

Factories: McKeesport, Pa. • Blairsville, Pa. • Pittsburgh, Pa.
Mt. Vernon, Ill. • Newark, N. J. • New Brunswick, N. J.

H. K. PORTER "Better Built" Equipment Established 1866

executive responsibility for one of the large automotive organizations. Mr. Crawford began his career as a draftsman with the old American Motor Car Co. in 1906, and later was advanced to assistant chief engineer. In 1913 he joined Chalmers Motor Co., serving successively as mechanic, designer, and chief draftsman before becoming assistant chief engineer. He obtained his first position as chief engineer when he joined Allen Motor Co. in 1917, later serving also as manufacturing manager. As chief engineer of Auburn Automobile Co. five years hence, he was responsible for the design of that car during the period of its rise. In 1927 he came to Chevrolet as assistant chief engineer and in two years was named chief engineer—the position he has held ever since. As chief engineer he has had an opportunity to utilize his ability to assign responsibility, follow through for results, and appreciate achievements of his associates. Added administrative responsibilities have not lessened his deep personal interest in and knowledge of detailed phases of technical development. He has also devoted considerable effort to cooperative engineering work of benefit to the industry and the profession as a whole.

♦

S. C. Hoey, a one-time machinist apprentice with Westinghouse Electric & Mfg. Co., has been appointed assistant director of the headquarters manufacturing engineering department.

♦

Linwood F. Millen has been named senior engineer, B-29 fuel pumps, Eclipse Machine Division, Bendix Aviation Corp., Elmira, N. Y. Mr. Millen had been B-29 engineer coordinator, Bell Aircraft Corp., Marietta, Ga.

♦

Dr. Howard W. Barlow who had been head of the department of aeronautical engineering, Texas A. and M. College, has been elected dean of the school of engineering.

♦

Philip A. Sidell has been appointed chief engineer of Gale Products Division, Outboard Marine & Mfg. Co., Galesburg, Ill. He had been chief engineer of DeBothezat Division, American Machine & Metals Inc., East Moline, Ill., of which company he still remains consulting engineer.

♦

Charles S. White has become chief engineer of Sav-Way Industries, Special Machine Division, Centerline, Mich. He formerly had been connected with Micromatic Hone Corp., Detroit, as research engineer.

♦

John Borland has joined the engineering staff of Clark Equipment Co., Buchanan, Mich. He had formerly been in the service engineering department of Elastic Stop Nut Corp., Newark, N. J.

♦

Soren H. Mortensen, chief electrical engineer, Allis-Chalmers Mfg. Co., Milwaukee, has been awarded the 1944 Lamme Medal of the American Institute of Electrical Engineers "for his pioneer work in the development of self-starting synchronous motors and for his contributi to the development of large hydraulic and steam-turbi driven generators."

♦

Rowland C. Lee, formerly assistant chief engineer Harvill Corp., Los Angeles, is now aerodynamics engine Northrop Aircraft Inc., Hawthorne, Calif.

♦

Vern Z. Perry, previously passenger bus body desig and production planner, Ward Body Co., Conway, A has been made president and general manager of Knigh town National Body Mfg. Co., Knightstown, Ind.

♦

Ezra Hollister has become connected with the Ben Aviation Corp., Pacific Division, Los Angeles, as hydrau development engineer. He left Jesco Co. where he h been chief engineer.

♦

Tom Elleman has joined Continental Motors Cor Muskegon, Mich., as engineering department design e gineer. He had been chief engineer, Bolens Products C Port Washington, Wis.

♦

Robert W. Reidis is now chief test engineer, resear and development, American Engineering Co., Philad phia. He had been test engineer, research and develo ment department, Rogers Diesel & Aircraft Corp., N York.

♦

L. L. Aspelin, formerly chief engineer, Romec Pu Co., has joined Thompson Products Inc., Cleveland, as sistant chief development engineer.

♦

David Prince, vice president of General Electric C has been placed in charge of the company's General E gineering Laboratory, the activities of which will be broa ened to include the requirements of the entire compan

♦

James M. Fiske, previously research engineer in char of laboratories, Aircooled Motors Corp., Syracuse, N. has been named mechanical engineer in the research d partment, Florida Division, Food Machinery Cor Dunedin, Fla.

♦

C. Robley Patterson has joined the Standard Oil C of Calif., El Segundo, Calif., in its mechanical engine ing design department. Formerly he had been mechanic engineer and manager of the Washington Office of Ligh Inc.

♦

William G. Newman, has recently become connect with Graham-Paige Motors Inc., Detroit, as engineer charge of experimental work. Prior to his new appoin ment he had been design engineer, Pioneer Engine & M Co., Detroit.

relays
IN MARINE COMMUNICATIONS

From ship to ship and from ship to shore—whether on war craft or on peacetime boats of commerce and travel—marine radio communications equipment plays a major role. Leading manufacturers of such equipment use Relays by Guardian, two of which are shown installed in the DC power supply unit of the HT-11 Radiophone manufactured by the Hallicrafters Company, Chicago.

Hallicrafters HT-11 Radiophone Unit Showing DC Power Supply

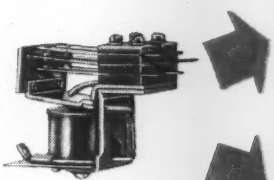
for Automatic Control of Electrical Circuits...

THERE'S A *Relay* BY GUARDIAN

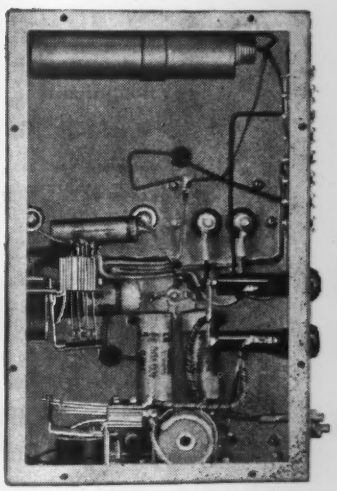
In this application one Guardian relay in its normal position feeds the input of the Vibrapack for receiving purposes. On the changeover from receiving to transmitting it disconnects the Vibrapack and simultaneously energizes the other relay. This in turn connects the Dynamotor input and output circuits.

Both relays are Guardian Series 115 with double wound coils for operation on 6 or 12 volts D.C. with the 6 volt winding in parallel and the 12 volt winding in series. It is a small, compact relay, ideal for use where space is limited.

Its use in Marine Radiotelephone is but one illustration of the many applications of relays in radio and electronic equipment. For complete description of numerous types of Relays by Guardian, write for Guardian's new Catalog No. 10.

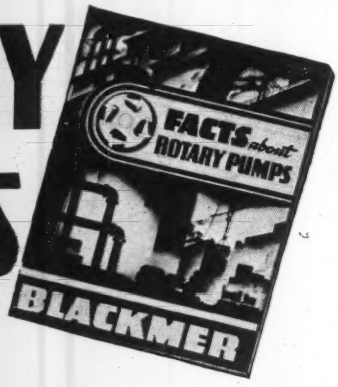


Series 115 DC Relay



GUARDIAN ELECTRIC
1601-D W. WALNUT STREET CHICAGO 12, ILLINOIS
A COMPLETE LINE OF RELAYS SERVING AMERICAN WAR INDUSTRY

FACTS about ROTARY PUMPS for



DESIGNERS OF MACHINES and PROCESSING EQUIPMENT

Alcohols
Acids
Beer
Butane
Dyes
Fuel Oil
Gasoline
Glue
Mash
Mayonnaise

Asphalt

Requiring PUMPS FOR THESE OR SIMILAR LIQUIDS

Tomatoes

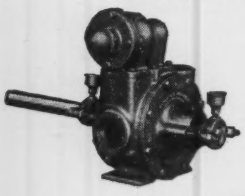
Molasses
Oils
Paints
Rosin
Soaps
Solvents
Soups
Syrups
Tar
Yeast

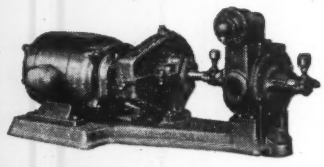
BLACKMER ROTARIES

are SELF-ADJUSTING FOR WEAR

"Bucket Design" swinging vanes automatically compensate for wear.

THIS MEANS SUSTAINED CAPACITY
DEPENDABLE OPERATION
AND LOWER PUMPING COSTS





WRITE FOR
Bulletin No. 306—Facts about Rotary Pumps
Bulletin No. 302—Pump Engineering Data

BLACKMER PUMP COMPANY
1970 Century Avenue Grand Rapids 9, Michigan

POWER PUMPS · HAND PUMPS · EZY-KLEEN STRAINERS ·

400-Cycle Aircraft Auxiliaries

(*Concluded from Page* 130)

from the auxiliary 28-volt bus.

Designers will not be entirely satisfied with this rangement. Although it saves considerable weight, p ably even more can be saved by eliminating the mo generator altogether and controlling the effective vol at the motor armatures by thyratrons. Still another met of controlling speed for turrets and for general app tions presupposes that the hydraulic variable-ratio d will be developed; it will then merely be necessary to terpose a small version of this transmission between a stant-speed motor and the device to be driven.

Many occasions arise for synchronized operation in or more parts of the airplane. For example, corresp ing wing flaps on both wings must be kept in step wi close limits or the airplane will become uncontroll For most applications, ordinary low-slip induction mo will give adequate synchronization, especially if the u are forced into exact coincidence at one or both end the cycle. In any case, the common system freque may be used by the ingenious designer as a reference for many synchronization schemes.

Will Be Fertile Field

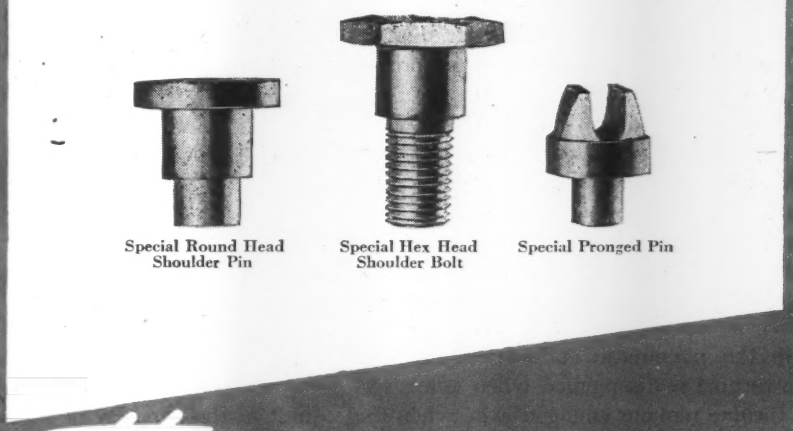
Designers will find a fertile field for other new dev which the alternating-current power system will make sible. There will also be many applications for such w known devices as transformers and static rectifiers. persil is extensively used in aircraft transformers and made possible savings in weight because of its magn properties at high frequencies and high densities.

Finally, the alternating-current equipment proper is from completely developed. Fame and fortune await designer who finally develops the variable-ratio dr Much work remains to be done on speed-governing me anisms even after the drive is available. Possibly an tirely new approach will solve the difficulty; for exam an electronic frequency converter which will convert v able-frequency alternating current to constant-freque

Methods of providing the necessary direct current probably less satisfactory than some future methods be. Static rectifiers have little overload capacity and instantaneous reserve power. Voltage regulators for th are either heavy or sluggish in operation. Motor-gen tor sets avoid these difficulties, but have high inputs light loads and require the maintenance normal to rota equipment.

Fault protection is being studied seriously as a resul combat experience. Really selective fault-clearing dev are needed, especially for military airplanes; these dev should clear damaged circuits almost instantaneously, never take out an undamaged circuit, or take out a cir so far from the fault that undamaged branch circuits disconnected.

The author acknowledges the assistance of H. J. Braun, H Keneipp and C. G. Veinott, all of the Aviation Engineering partment, Lima Works, Westinghouse Electric & Manufa ing Co., particularly for furnishing curves and other basic upon which this article is based.



Upsetting AUTOMOTIVE PARTS

SPEEDS PRODUCTION

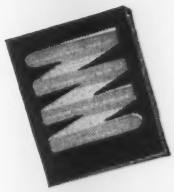
CUTS COSTS

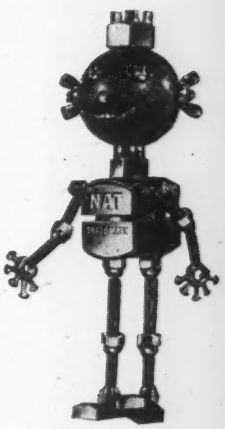
Here are three special parts used in large quantities (two of them run into millions) by a large manufacturer of automotive equipment. They have both war and peacetime application.

They were originally made milled from bar. When sufficient production could not be obtained by that method, the problem was put up to *National*. We developed an upsetting procedure that met the need for *volume production* and also resulted in substantial *cost reductions*.

You may want to look into the possibilities of saving time and money on fasteners, for present needs or future. Or you may have a problem of fastener improvement on which our engineers can help you as they have many other manufacturers.

Chances are you'll be surprised at what can be done.





Send for a copy of this case history booklet describing briefly, with diagrams, 15 typical instances of time, materials and money saved. Ask for the "Savings" booklet.

National
HEADED AND THREADED
PRODUCTS

THE NATIONAL SCREW & MFG. CO., CLEVELAND 4, O.

Turbine Locomotive

(*Concluded from Page* 142)

for each of the inlet pipes, control steam admission to the forward turbine, opening the valves in sequence.

Reverse turbine has a single Curtis stage and is overhung on an extension of the reverse pinion shaft, *Fig.* 2. Although the small reverse turbine uses only one-third of the steam flow of the forward turbine, tractive effort in reverse is almost as high due to the 4:1 reduction gear between the reverse turbine and the high-speed main pinion. Maximum locomotive speed in reverse is 22 miles per hour, turbine speed about 8300 revolutions per minute.

Power in reverse is transmitted to the main high-speed pinion through a hydraulically actuated positive-engagement clutch. Engagement or disengagement of the clutch while the locomotive is moving is prevented by a "zero-speed" interlock in the pneumatic control circuit. Although the reverse turbine is disengaged when going forward, the forward turbine remains engaged even when the locomotive is traveling in reverse.

A single lever controls both speed and direction of the locomotive. The control consists of a forward "pneudyne" (an air relay-controlled pneumatic cylinder), a reverse pneudyne, overspeed and low-oil-pressure protection valves and the zero-speed interlock. The pneudynes rotate the throttle valve camshafts, controlling steam to the turbines.

As experience is gained with the geared turbine locomotive, it seems highly probable that even larger locomotives of this type will be built. Although the ultimate

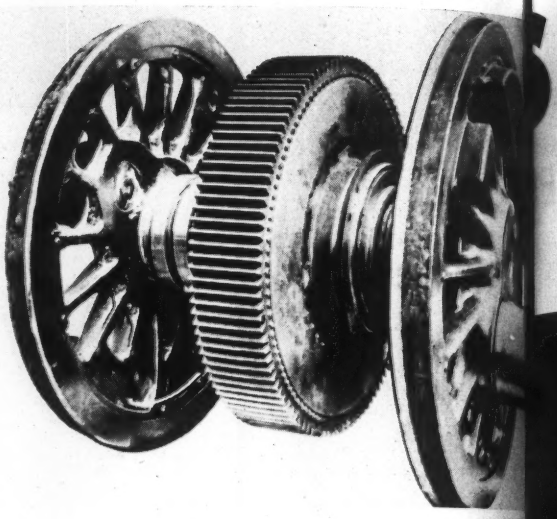
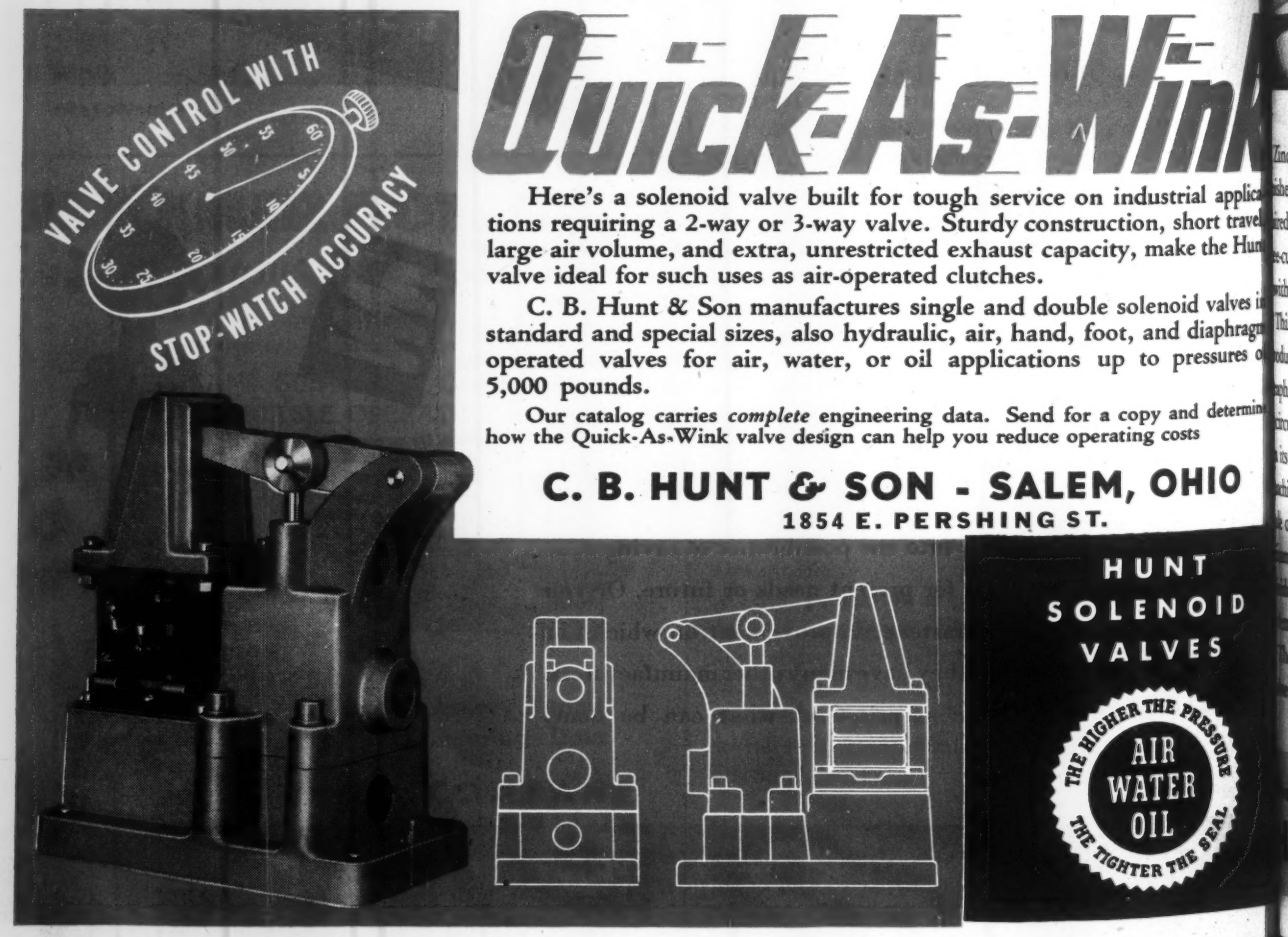


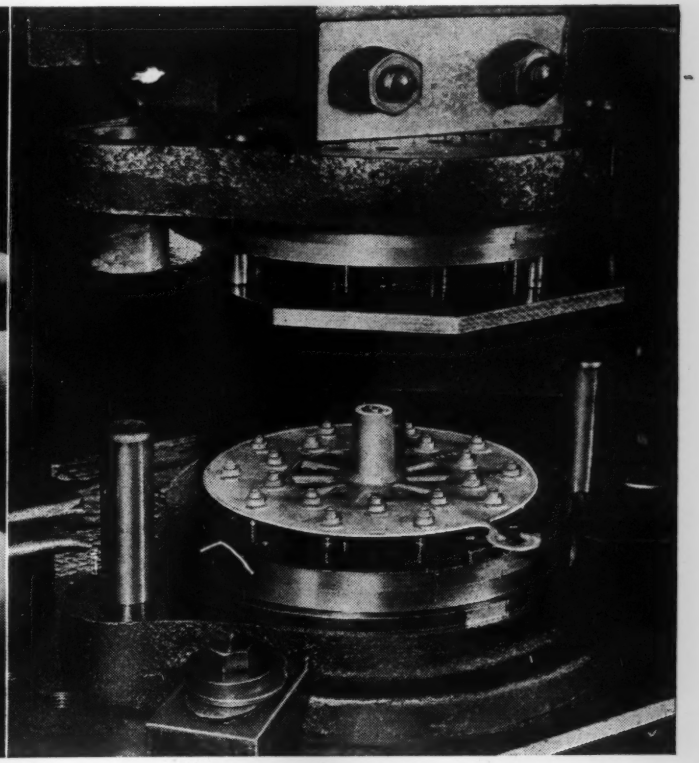
Fig. 3—Low-speed gear drives main axle through drive" which permits relative motion of axle and

limit in the capacity of the steam locomotive is gove by boiler dimensions, which in turn are limited by fa such as clearance and visibility, the geared turbine with its smaller driving wheels and shorter wheelbase mits a somewhat larger boiler and more grate area u the boiler. Further, the turbine can utilize steam at peratures prohibitive with reciprocating engines, with sequent improvement in efficiency. It seems likely, t fore, that geared turbine locomotives of 10,000 horsep will eventually be available for accelerated passenger freight schedules of American railroads.


VALVE CONTROL WITH
STOP-WATCH ACCURACY
Quick-As-Wink
Here's a solenoid valve built for tough service on industrial applica
tions requiring a 2-way or 3-way valve. Sturdy construction, short travel
large air volume, and extra, unrestricted exhaust capacity, make the Hun
valve ideal for such uses as air-operated clutches.
C. B. Hunt & Son manufactures single and double solenoid valves i
standard and special sizes, also hydraulic, air, hand, foot, and diaphragm
operated valves for air, water, or oil applications up to pressures o
5,000 pounds.
Our catalog carries complete engineering data. Send for a copy and determin
how the Quick-As-Wink valve design can help you reduce operating costs
C. B. HUNT & SON - SALEM, OHIO
1854 E. PERSHING ST.
HUNT
SOLENOID
VALVES
THE HIGHER THE PRESSURE
AIR
WATER
OIL
THE TIGHTER THE SEAL

SHORT CUTS!

—MACHINING PROBLEMS SIMPLIFIED WITH ZINC ALLOY DIE CASTINGS



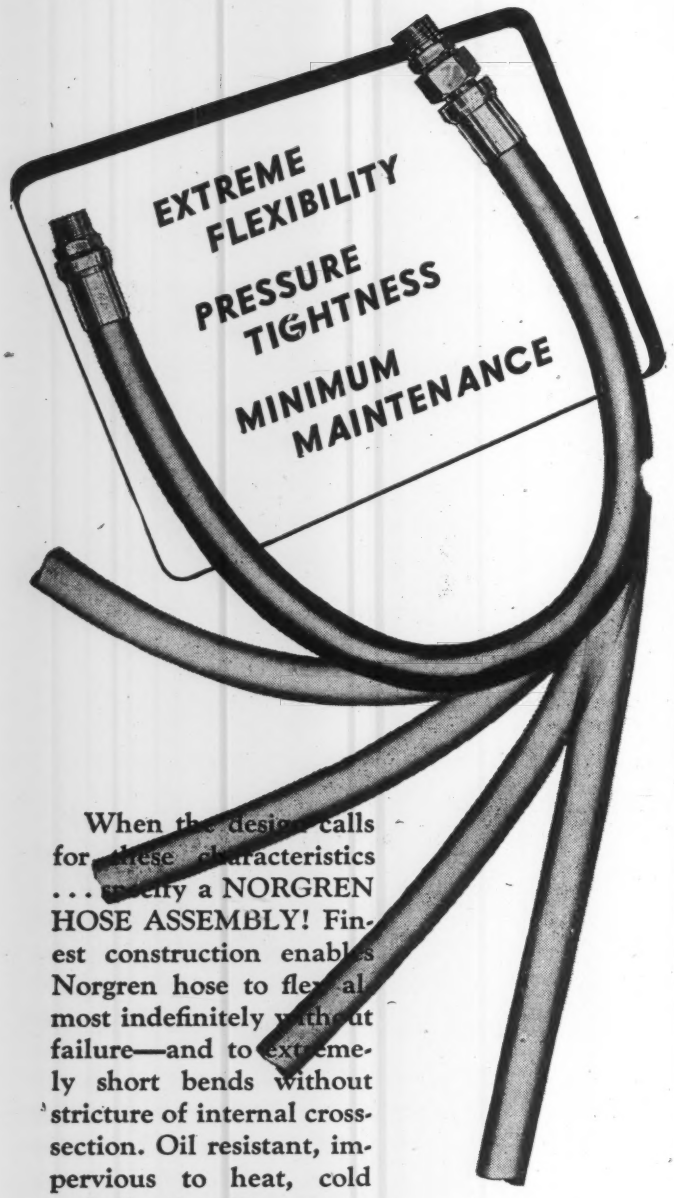
Zinc alloy die castings are often produced so close to ished dimensions that little or no machining is re- ired. When machining *is* called for, however, the e-cutting zinc alloys insure easy, rapid metal removal ith long tool life.

This short-cut factor is illustrated in the high speed oduction of a shell fuse casting (see inset in photo- aph directly above.) 20 of these fuse parts are cast on ircular gate in one shot—and this gate has hooks cast its outer edge for ease of conveying from the casting achine. Two locating pins are also cast on the under- e of the gate for accurate and rapid positioning in a ming die (right) where, in one punch, the 20 in- dual fuse castings are removed. The remaining gate melted and reused.

The free-cutting quality of zinc alloys facilitates the sequent drilling of a hole (see inset) in each casting and thus, **in just three operations—casting, trimming and drilling—the vitally needed shell fuse castings are rapidly produced.** It is the combination of high speed production and excellent physical characteristics which have made die castings of zinc alloy the most widely used. *Every die casting company is equipped to make zinc alloy die castings,* and will be glad to discuss their advantages with you—or write to The New Jersey Zinc Company, 160 Front Street, New York 7, N. Y.

ZINC
FOR DIE CASTING ALLOYS

Research was done, the Alloys were developed, and most Die Castings are specified with
HORSE HEAD SPECIAL (99.99+% Uniform Quality) ZINC



When the design calls for these characteristics . . . specify a NORGREN HOSE ASSEMBLY! Finest construction enables Norgren hose to flex almost indefinitely without failure—and to extremely short bends without stricture of internal cross-section. Oil resistant, impervious to heat, cold and vibration. Permanent machined couplings. Hose steel wire reinforced for high pressures. Replaceable couplings for low pressures. Full line covers all requirements, hydraulic or pneumatic.

Write for information, giving your specifications. C. A. Nogren Company, Denver 9, Colorado.

LIMBER AS A LARIAT STRONG AS STEEL

Norgren

BUSINESS AND SALES BRIEF

AMONG other appointments recently made by Foote Gear & Machine Corp., Chicago, is that of R. B. as assistant vice president in charge of sales engineerin the industrial gear division.

♦

With headquarters in St. Louis, R. L. Heath has jo Climax Molybdenum Co. in the capacity of metallurgical gineer. Previously he had been chief metallurgist of the lison Division of General Motors Corp. at Indianapolis.

♦

Parker-Kalon Corp., New York, has named S. S. Kah sales and advertising manager to succeed the late Ch S. Trott. Mr. Kahn had been assistant sales manager for teen years.

♦

Completion of a processing plant with 10,000 square of space has been announced by General Controls Inc., G dale, Calif. The plant will house the heat treating, ano ing, mechanical deburring, sand blasting and painting partments. Nearing completion is also a new modern o building.

♦

Formerly vice president and general manager, W. F. Lis has succeeded the late Thomas B. Fordham as president Leland Electric Co., Dayton, O.

♦

Wendell A. Melton has been named district manager charge of the Tulsa territory for The Foxboro Co., and make his headquarters at the company office located in McBirney building, Tulsa 3, Okla.

♦

According to a recent announcement, Doehler Die Cast Co. and W. B. Jarvis Co. have consolidated under the na of Doehler-Jarvis Corp.

♦

Recently announced by U. S. Gauge Co., Sellersville, Pa. the relocation of the Chicago office to the Monadnock Blo 53 Jackson boulevard. In charge as district manager will Walter H. Magee.

♦

Air Reduction Sales Co. has made the following appoi ments: H. F. Henriques, previously sales manager of the no central division, has been named general sales manager; J. Lincoln, who had been sales manager of the south cent division, has been made director of sales services; C. Bloodgood, formerly sales manager of the Pacific Coast divisi

BEHIND THE SCENES WITH THE MEN WHO PUT THE "KNOW-HOW" INTO PILOT MOTORS

How PILOT Motor Research Helps You Get The Right Motor For The Job!

Like properly fitted eye glasses, the *right* motor for a job must be carefully prescribed or specified and then produced according to the specifications.

In establishing production specifications, we have to learn, first of all, what the job requirement of the motor is—what the power drive application is to be . . . under what conditions the motor must operate . . . whether it is for continuous or intermittent duty, etc. Once this information is known, trial specifications are made and a motor built.

The test board (shown above) puts this sample motor through its paces to make sure it embodies the characteristics specified by our engineers in accordance with the customers' requirements. This test records motor speed, power, heat rise, current required, work performed—both running free and under full load. In short, it determines over-all motor efficiency.

If necessary, details of construction are changed and another sample built and tested before final production specifications are released. During manufacture, production pieces are tested and re-tested to make sure that the motors conform to the customer's needs. This procedure, followed in specifying and producing "custom-built" PILOT Shaded Pole Motors, is assurance that you are getting the *right* motor for the job.

Because PILOT Motors, too, have gone to war, production for civilian and postwar needs has necessarily been curtailed. High priority orders, however, are being accepted according to our ability to produce and deliver; and postwar applications are considered as promptly as possible.

F. A. SMITH MANUFACTURING CO., INC.
401 DAVIS ST. ROCHESTER 2, N. Y.

FASCO

Pilot Shaded-Pole MOTORS & BLOWERS

has been appointed assistant to the vice president in charge of sales. All these appointees will make their headquarters at the main office of the company located at 60 East Forty-second street, New York. H. P. Etter, who has been named to succeed Mr. Bloodgood as sales manager of the Pacific Coast division, will make his headquarters in the Mills Tower building, San Francisco. Mr. Etter had been manager of the Los Angeles district.

♦

Opening of a new regional office at Lansing, Mich., has been announced by Twin Disc Clutch Co., with G. M. Guilbert as manager. The new office will better serve the many Michigan manufacturers who had formerly been served from the Cleveland territory.

♦

With manager L. W. O'Day in charge, the Peoria, Ill., office of The Lincoln Electric Co. has been moved to larger quarters in the Electrical building, 214 Second street.

♦

Graver Tank & Mfg. Co. Inc., East Chicago, Ind., has announced the appointment of Charles W. Springer as manager of eastern sales, with offices at 424 Madison avenue, New York 17

♦

Change in the name of Rockford Drilling Machine division of Borg-Warner Corp., at Rockford, Ill., to Rockford Clutch division, was recently made known by the corporation.

♦

Among several changes in the organization of the apparatus department is the following which has been announced by General Electric Co.: H. V. Erben, previously commercial vice president and manager of the Central Station divisi has been appointed assistant general manager of the appar department.

♦

Offices at 150 Broadway, New York, have been establis by Associated Engineering & Research Companies, an org ization of inventors, engineers, research men and market alysts serving manufacturers seeking new products in electro-mechanical field.

♦

Moving to its new factory at 4114 North Knox avenue, C cago 41, has recently been announced by Mead Specialties

♦

Stanley & Patterson Division of Faraday Electric Co has moved to 434 Newbury street, Boston 15. Combined this address will be the machinery and personnel of Holtzer-Cabot Signal division as well as equipment and p sonnel from the Faraday plant at Adrian, Mich. Paul E. F wald has been named sales manager of this division.

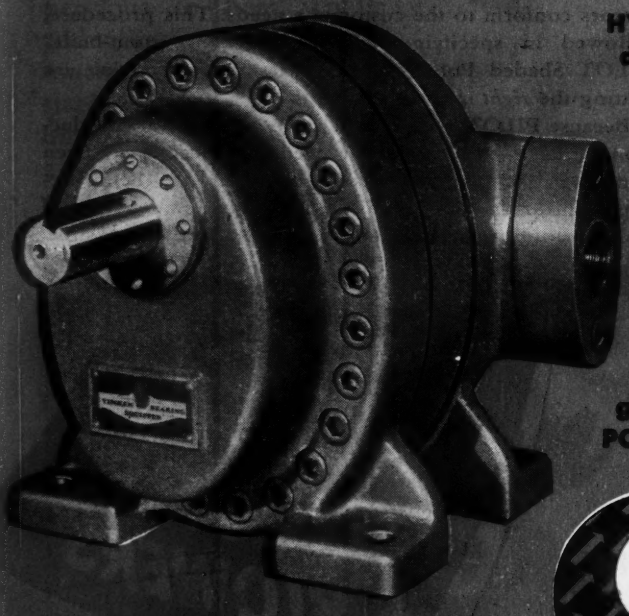
♦

According to a recent announcement, Precision Plastics Philadelphia, has moved from 1724 West Indiana avenue 4655 Stenton avenue.

♦

CORRECTION—L. V. ("Larry") Nagle has been named v president in charge of national sales for Udylite Corp., D troit. It was incorrectly stated in the February issue t L. K. Lindahl, president of the corporation, had been pointed to this position.

HYDRO-POWER
GEAR PUMPS for Oil Hydraulic Circuits



HYDRO-POWER gear pumps are constant delivery pumps designed for generating pressure up to 1000 lbs. per sq. inch. Close tolerances are maintained between working parts. An exclusive gear design reduces the hydraulic load on the gears. Precision tapered roller bearing mounting of gear shafts accurately positions pump gears, with close tolerance between gears and housings. Wear is reduced to a minimum. Adjustments are provided for compensation for bearing wear. Available in capacities from 5 to 60 gallons per minute. Write today for HYDRO-POWER bulletin 440.

hydro power

HYDRO-POWER

HYDRAULIC PUMPS AND CONTROLS - VALVES - CYLINDER AND RAM ASSEMBLIES - POWER UNITS - SYSTEMS - SPECIAL HYDRAULIC EQUIPMENT

HYDRO-POWER SYSTEMS, INC.

Mount Gilead, Ohio, U.S.A.

Only molybdenum in cast iron improves both strength and toughness

LIMAX FURNISHES AUTHORITATIVE ENGINEERING DATA ON MOLYBDENUM APPLICATIONS.

ARMY E NAVY

MOLYBDIC OXIDE, BRIQUETTED OR CANNED • FERROMOLYBDENUM • "CALCIUM MOLYBDATE"

Climax Molybdenum Company

MOLY

500 Fifth Avenue · New York City